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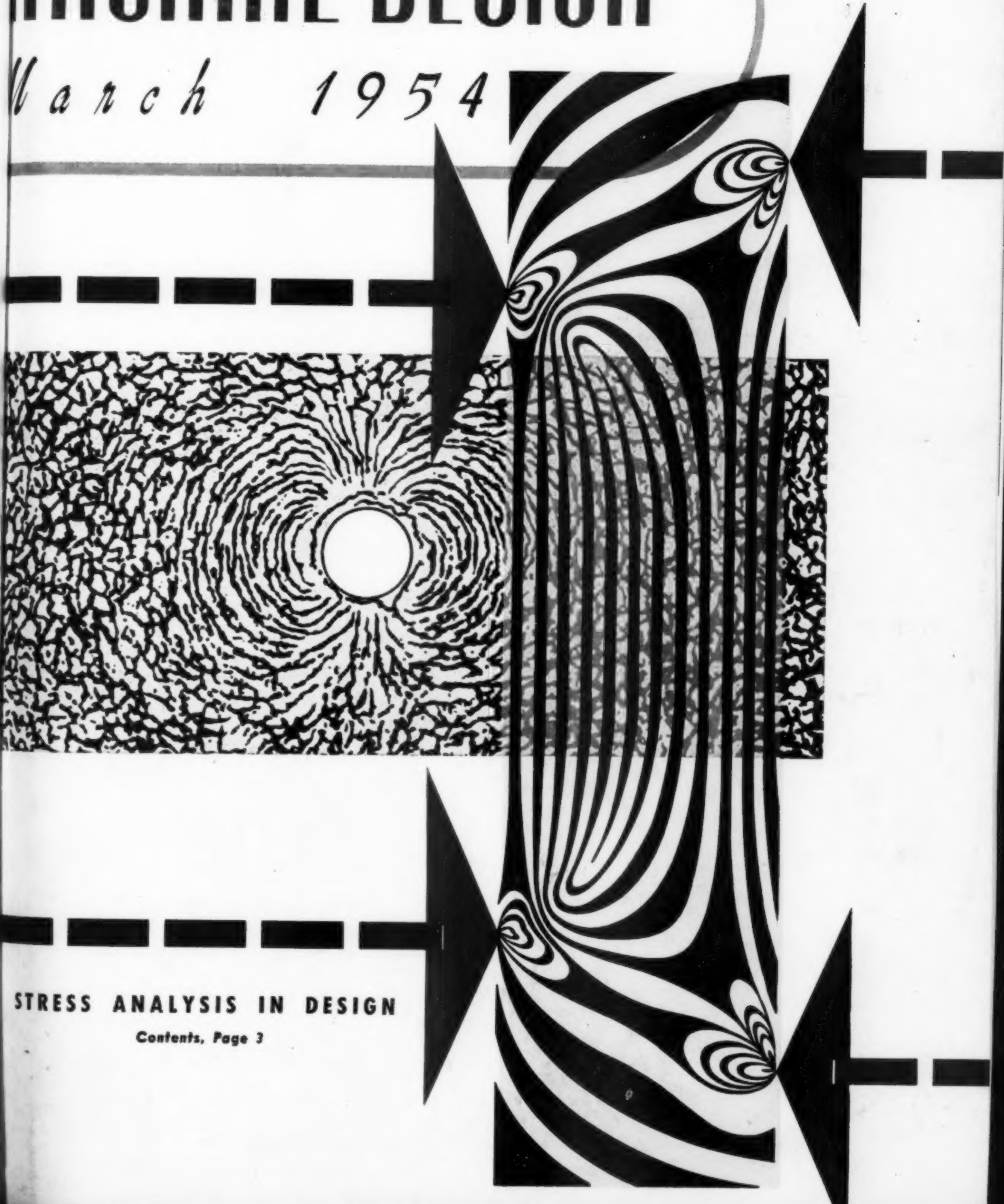
MACHINE DESIGN

March 1954



STRESS ANALYSIS IN DESIGN

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THE PROFESSIONAL JOURNAL
FOR ENGINEERS AND DESIGNERS

MACHINE DESIGN

MARCH 1954
Vol. 26—No. 3

REGULAR DEPARTMENTS

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Over the Board

John Holt

A new name appears on our masthead (the left-hand column on Page 3) this month—that of John Holt, assistant editor, whose picture appears on Page 321. He is one of the additions to the editorial staff made necessary by the transfer of Roger Bolz to *Automation*. An engineering graduate of Case Institute of Technology (1951), John comes to us from the Chesapeake & Ohio Railway where he was a design engineer in the research department. During his period of service with the army he was stationed in India, where he worked on maintenance of railroad mechanical equipment. A lifelong enthusiasm for railroading finds continuing satisfaction in his membership in the National Railway Historical Society and the Federation for Railway Progress, and in his daily commuting—on the Erie Railroad—to Cleveland. Like his colleague Bob Rodgers, John seems to the rest of us ordinary mortals to be bumping his head against the stratosphere—a six-footer plus.

This Month's Cover

Stress analysis, once regarded as the epitome of theoretical activity, has long since reached a highly practical plane, due in large part to the contributions of experimental stress analysts. With stress distributions beautifully pictured through such techniques as photoelasticity and brittle coatings, the practical engineer can see with his own eyes how his designs are likely to perform in service. Hartman and Benner's 4-part series of articles, commencing in this issue (Page 134), present a comprehensive view of how theoretical and experimental methods complement each other in practical design. To highlight the subject on this month's front cover, George Farnsworth has illustrated the photoelastic stress pattern in one typical load situation and the brittle coating pattern in another.

The Stork Is Coming

We're expecting an addition to the Penton family this year. Our baby sister is to be named *Automation* and the obstetricians will be Roger W. Bolz and Edward S. Lawson. Roger (see Page 318) will be editor of the new publication and Ed, formerly New England district manager of MACHINE DESIGN, will be business manager. Inasmuch as no existing publication, including MACHINE DESIGN, has space to do full justice to the specific needs of the automation field—the science of automatic production — the new magazine will fill in a unique manner a rapidly growing demand. Incidentally, although Roger will no longer be active on the editorial staff of MACHINE DESIGN, you will still see his name listed on our masthead as contributing editor and, from time to time, as author of articles in our pages.

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CCA NBP

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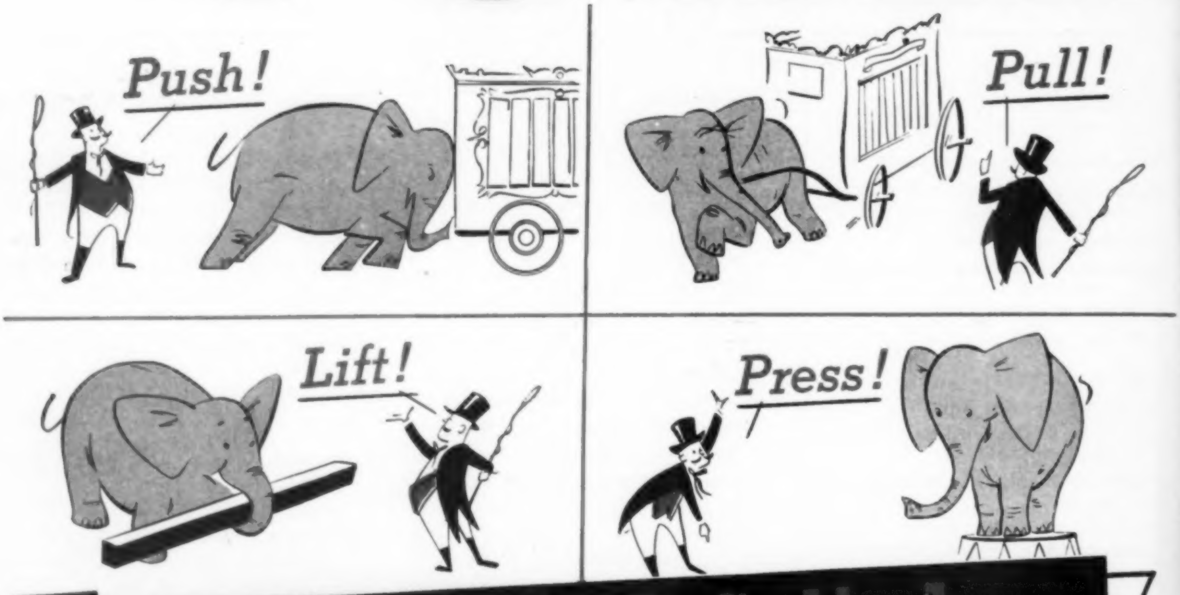
(*Concluded on Page* 10)

get controlled POWER



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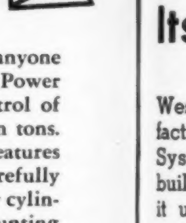

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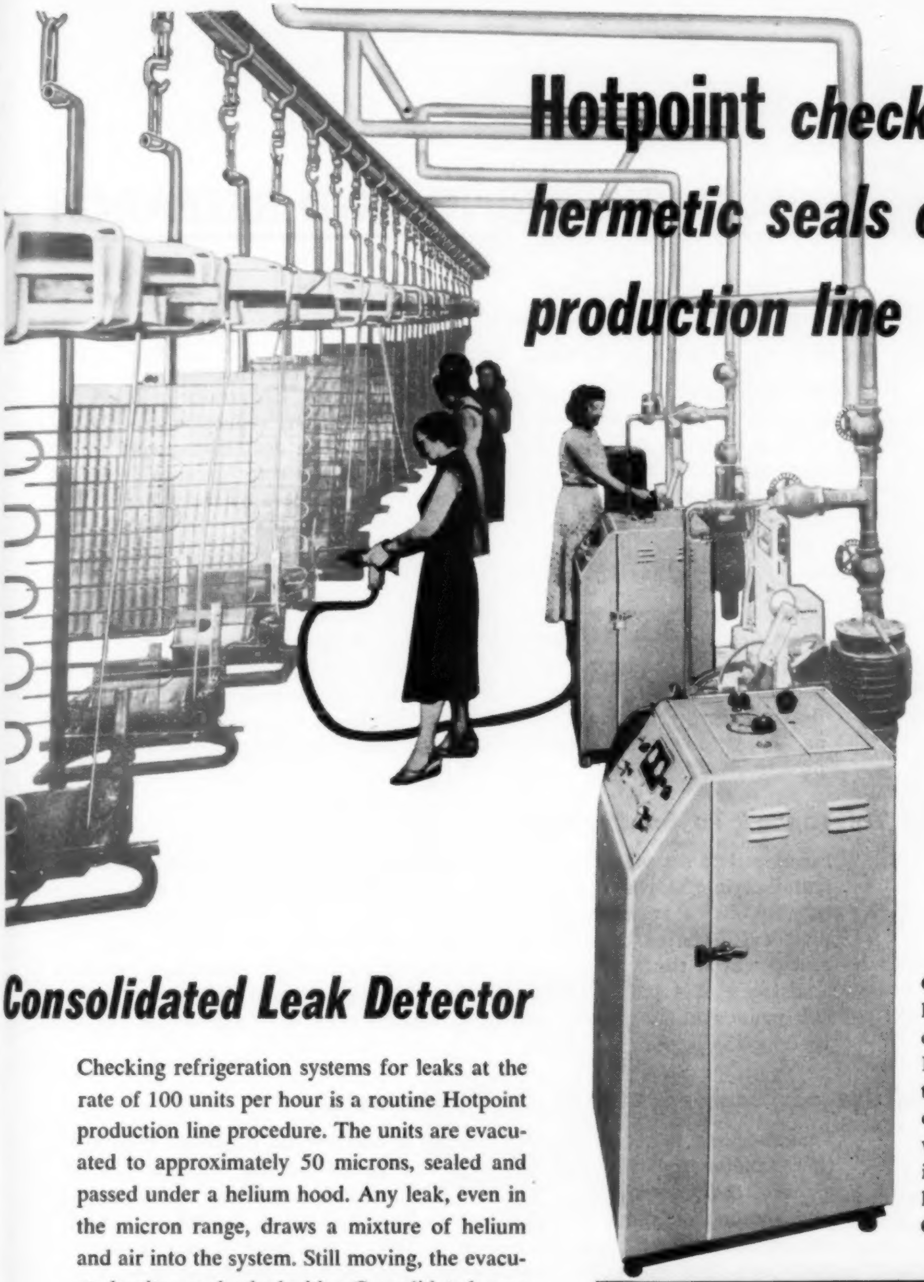
MACHINE DESIGN is indexed in Industrial Arts and Engineering Index Service, both available in libraries, generally

Hotpoint *checks hermetic seals on moving production line*



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Checking refrigeration systems for leaks at the rate of 100 units per hour is a routine Hotpoint production line procedure. The units are evacuated to approximately 50 microns, sealed and passed under a helium hood. Any leak, even in the micron range, draws a mixture of helium and air into the system. Still moving, the evacuated units are checked with a Consolidated mass spectrometer-type leak detector. The slightest trace of helium sounds an audible alarm and the faulty unit is pulled off the line. These mobile, highly sensitive leak detectors are adaptable to a wide range of operations for production line control of both vacuum and pressure systems.

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MACHINE DESIGN

Topics ... IN ENGINEERING AND RESEARCH

Cold Reduction Increases Titanium Creep Resistance

Creep testing of titanium and two of its alloys by University of Michigan for the Air Force indicates that a small degree of reduction greatly increases creep resistance. Reduction of about 7 per cent during cold working seems optimum; further cold working continues to increase creep resistance but at a decreasing rate. Data were collected by reheating at 210, 400 and 600 F.

Short-Radius Grooves Produced with Wire

Short-radius grooves in metal blocks, such as those required for magnetic wire recording heads, can be produced by using stainless steel wire as an abrasive. According to Fort Wayne Metals Inc., the wire surface is oxidized to varying degrees to produce the abrasive properties desired. When drawn across the metal block, the wire forms a groove of the same radius as the wire—and is said to give better results than any other known procedure.

Human "Jiggle Tolerance" To Be Explored

Developed to set vibration tolerance in design of new equipment, a machine installed in the Naval Medical Center in Bethesda, Md., will explore the human body's reaction to constant vibration. The device consists of a platform actuated by a 25-horsepower motor which can jiggle a man up and down 50 times a second or, at lower frequencies, bounce him through a distance of 4 inches. The machine is designed for a maximum load of 200 pounds at any combination of displacements and frequencies not exceeding 15g peak acceleration.

First Successful Use of Tungsten Resistance Thermometers

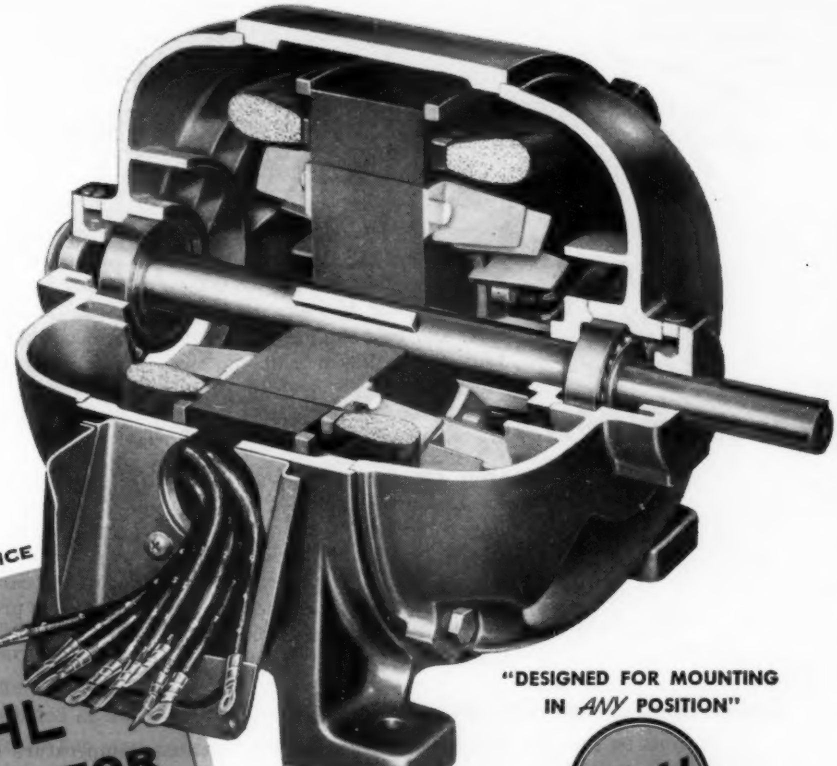
Tungsten has been successfully used in a commercial precision resistance thermometer for the first time. According to General Electric Co. engineers, the present "best" metal for this use is platinum, which has an upper range of about 600 C. Present tungsten rating is 300 C, but excellent stability at temperatures up to 1000 C has been shown, and forecasts are that the upper temperature limit will be increased during the next few years.

Silver Plating—Without Electricity

Applied by swabbing on with a soft cloth, a new silver plating solution is said to produce a silver plating without electricity, and is being used for plating connections, terminals and electrical joints to increase conductivity and reduce resistance. Two solutions are available from Specialty Manufacturers and Distributors, one for nonferrous metals such as silver, copper, brass and bronze, and the other for nonferrous and other metals.

Load Calibration Devices Accurate to 12 Million Pounds

Recently completed, four 3-million pound capacity compression dynamometers can measure forces up to 6000 tons accurately. Developed by the National Bureau of Standards, the dynamometers use wire resistance strain



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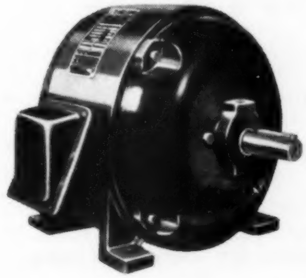

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gages as part of their load indicating systems. Previous high capacity was 2.5 million pounds, obtained by the simultaneous use of a number of devices of 200,000 and 300,000-pound capacity. It is estimated that testing machines can be calibrated to 10 million pounds in about one-fourth the time and half the cost formerly required to calibrate to 2.5 million pounds. Largest testing machine known is a 10 million pound unit in the NBS laboratories.

25 Billion Volt Atomic Accelerator To Be Built

Over ten times more powerful than the largest atomic accelerator now in existence, a new alternating-gradient synchrotron to be built at Brookhaven National Laboratory will produce proton beams of energies ranging up to 25 billion electron volts. Most powerful accelerator now in existence is the Brookhaven Cosmotron, which can accelerate protons to 2.3 electron volts. Also under construction at the present time is the Bevatron at University of California, which is expected to be used in the 5 to 7 billion electron volt range.

Superior Optical Cement Developed

Capable of withstanding extreme temperature variations, a new optical cement—plasticized cellulose caprate—appears suitable for replacement of Canada balsam in all types of optical instruments, according to the Navy. The synthetic resin is an improvement over the former material, which has been standard in the Navy for airborne optical instruments, in that it is sufficiently fluid to be cemented at a temperature of less than 250 F, permitting the cementing of large camera lenses.

Electroformed Molybdenum Now a Possibility

Electroformed parts of molybdenum, and more rapid, less expensive methods for producing parts and coatings, may be possible through a new electrolytic preparation method. Developed by Drs. Seymour Senderoff and Abner Brenner of National Bureau of Standards, the method has produced high-purity molybdenum by electrolysis of a bath of fused salts. Flexible and easily controlled, the process can form deposits varying from fine powders to thick, coherent layers.

Voltage Displacement Melts Ice on Power Lines

A relatively new procedure — angular displacement of voltage — can heat up power lines sufficiently to melt ice, according to O. L. Oehlwain of Public Service Co., Chicago. By choice of line length and voltage, the additional circulating current imposed plus normal load current can be made sufficient either to melt or prevent formation of ice. Voltage along the line section varies from normal at the extremities to a minimum at the electrical center.

TV Pictures and Sound Recorded on Magnetic Tape

Still in the development stage, a system for recording black and white or color TV picture and sound on magnetic tape has been developed by Radio Corp. of America. Using principles similar to audio recording, the TV recording system involves much greater problems; audio signals run from 20 to 20,000 cps, but video signals range up to 4 million cps. The system puts five parallel channels on half-inch tape—three for primary color, one for sound, and one for synchronizing. Tape speed is a high 30 feet per second, giving 4 minutes of recording time on a 17-inch diameter reel.

MACHINE DESIGN
MARCH 1954

Are Engineers Professional?

TWO recent reports published by the National Society of Professional Engineers contain much food for thought by management and by engineering employees alike. The first, titled *Professional Engineers' Income and Salary Survey,* brings out among many other facts that fifty per cent of mechanical engineers in 1952 earned $8250 or more. This figure is also the median for mechanicals with 14 or 15 years of job experience. Corresponding salary for two years' experience is $5000 and for 30 to 35 years' experience, $10,000.

These median figures do not, of course, tell the whole story. Ten per cent of mechanical engineers earned at least $18,720 and 25 per cent at least $11,860. On the other hand, another 25 per cent earned no more than $6420 and 10 per cent no more than $5240. For electrical engineers the median figure was $7940 and for all engineers, $7850.

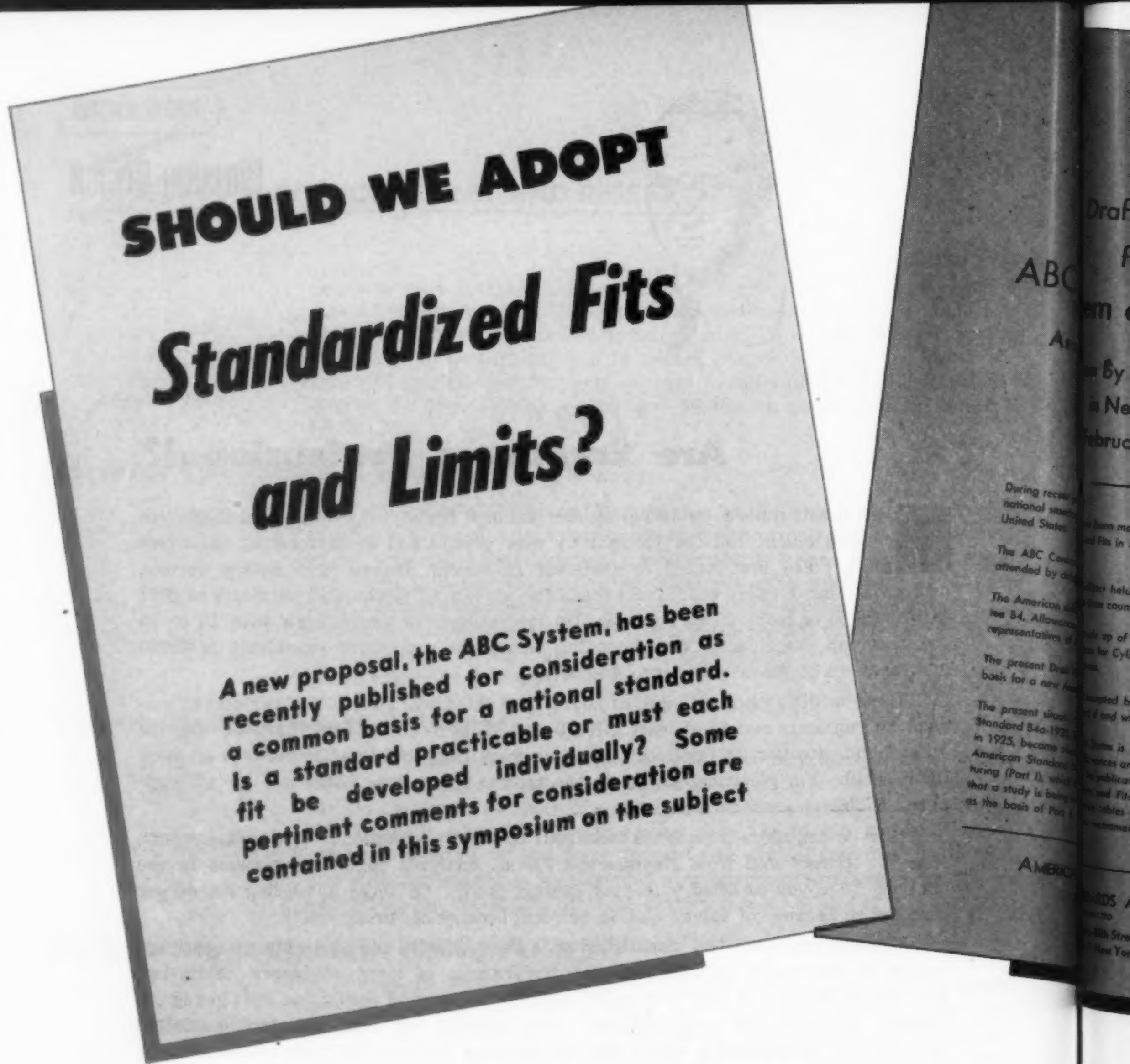
How do engineers themselves feel about the salary situation? The second report, *How To Attract and Hold Engineering Talent,* analyzes engineers' answers to the question "Are you satisfied with your present job?" Of those answering no, 45 per cent did so because of salary and 34 per cent because of "prospects."

If many engineers feel dissatisfied with their salaries and prospects, management in turn is not wholly satisfied with the performance of many engineers. Sixty-two per cent of executives who were questioned for the second report feel that young engineers' attitudes are unprofessional. One pointed out that too many engineers fell they can understand a bigger job but overlook the fact that their output on their present job is far from satisfactory. Many fail to appreciate the relationship between time and dollars.

What can be done to improve the mutual relationships of engineers and their employers? The answer seems to lie in the word "professional." Here management can help stimulate professional attitudes by clearly differentiating between engineers and nonprofessionals in personnel policy and by exposing engineers to situations that demand the professional approach. Active encouragement to engineers to become licensed and to join their professional societies can also help. That 43 per cent of employers now pay initiation fees and dues in professional societies for at least some of their engineering employees is a healthy sign.

As for engineers, if current salaries and advancement prospects seem to leave something to be desired, they might well look into their own attitudes and conduct on the job before giving vent to their gripes. It should never be forgotten that professional recognition of engineers depends on engineers acting like professionals. Only when enough engineers do so can the profession as a whole expect to achieve the recognition that some individuals already have earned.

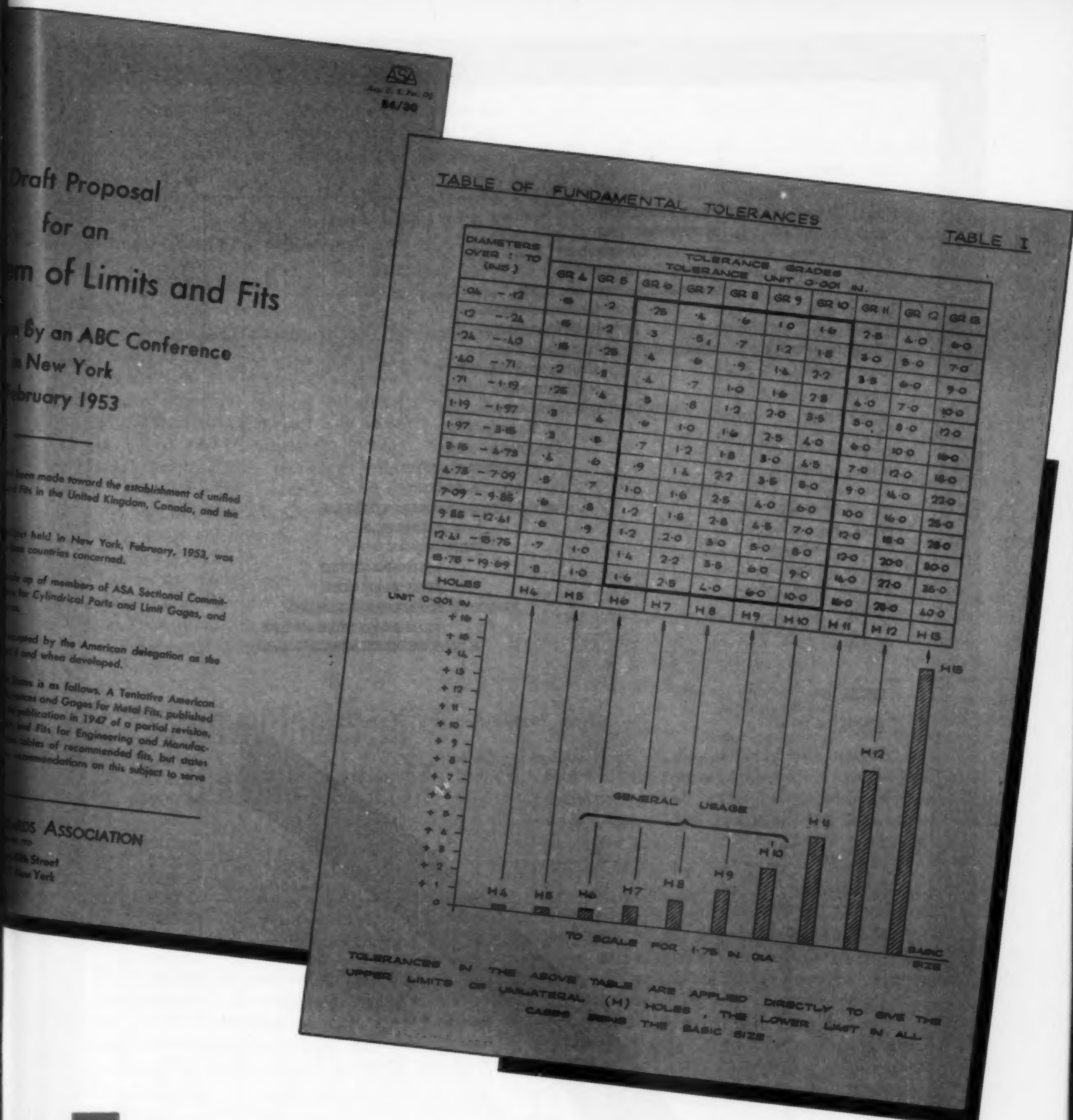
Colin Carmichael
EDITOR



WITHOUT question, it behooves engineering management to fully appraise and employ standards to the greatest extent possible in design. The tremendous value of simple standards such as those governing threads and fasteners needs little comment. But today, the wide variety and extent of standardization require more than passing attention. Few engineering departments are aware of the many areas now covered by standards or are capitalizing on the dividends in time and money possible.

One area of standardization interest of particular note today is that of fits and limits. In the past the ASA Standard B4a on Limits and Fits received little attention, was little used and was finally abandoned. Renewed interest has brought about a new proposal to set up a standard in conjunction with Canada and Britain. This standard, known as the ABC System of Limits and Fits ASA B4/30, is somewhat similar to the original ISA system but has been greatly improved. To assist in publicizing this new proposal and to attempt to define the general attitude of industry to such a standard, MACHINE DESIGN has asked many engineers for their considered comments regarding it. This symposium will define by such individual comment some of the pros and cons. It is hoped that perhaps such thoughtful consideration may lead to a greater awareness of the problem, its practical aspects and a solution mutually satisfactory to most engineers. The economic and practical problems are great, but without a real effort, it appears that the desirable end result of adoption may not come to pass.

In surveying a wide cross section of industry wherein fits and limits cannot help but be critical, it seems that little or no consideration has been given this problem. Practices vary from none in particular to specific systems arranged for one plant or one industry. But most outstanding probably is the fact that, even today, the fundamentals of tolerancing for fits and limits, the basic systems available and the effect of fits and limits on cost remain little known and little understood.



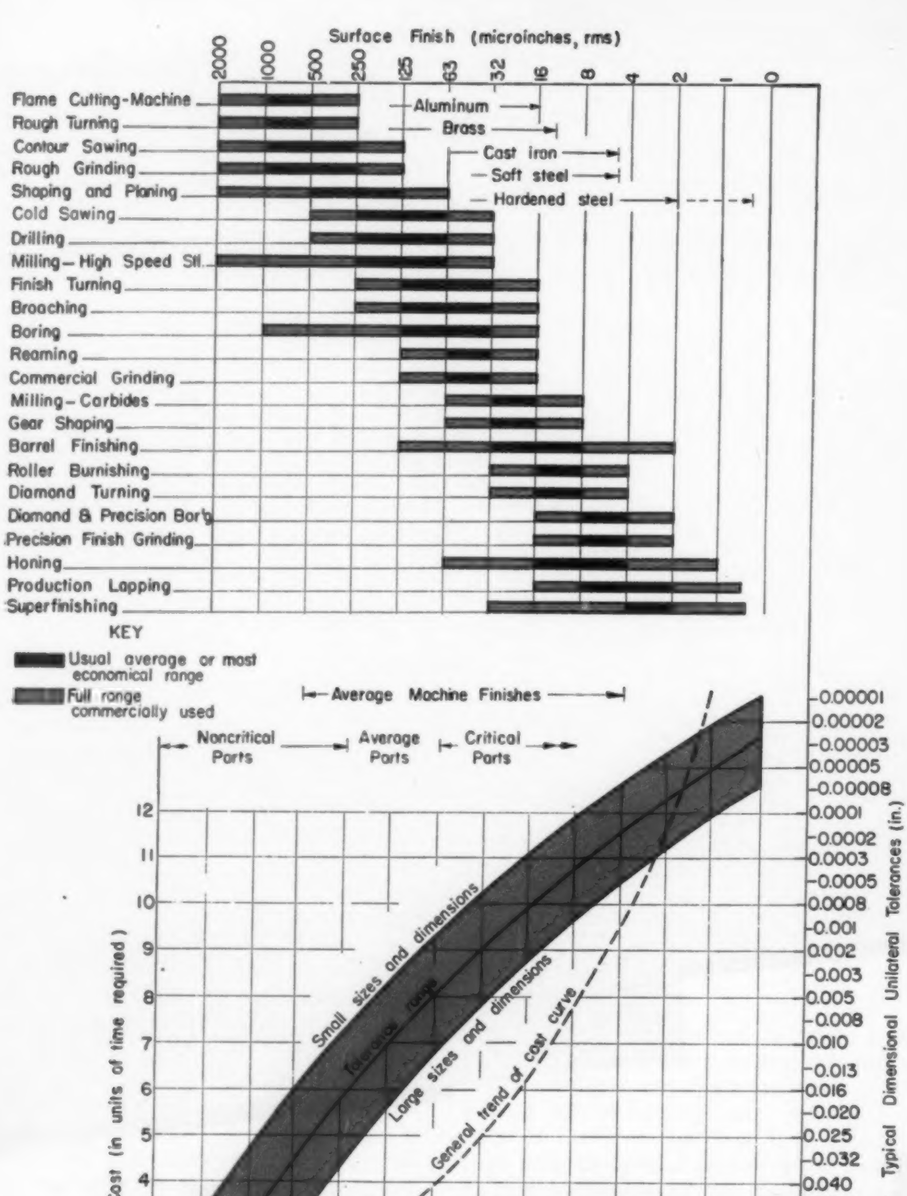
Why Not Simplify?

By Roger W. Bolz
Associate Editor, Machine Design

TO THE thoughtful engineer, there is little question as to the desirability or value of a standardized system for fits and limits. However, practically speaking, any system that merits adoption must meet certain requirements. It must be generally applicable throughout industry and it must be adaptable with minimum cost and difficulty.

Probably the major hurdle which prevented earlier adoption by industry of a standard has been complexity. It remains today that any system must be simple enough to permit use by average engineers without great study. There is little reason for complexity. Simplicity is the byword in design but it is never achieved by the easy path. Without doubt, a simplified system of fits can also be obtained by dint of hard work.

Because the real economy in design derives from



Surface Finish (microinches, rms)
2000
1000
500
250
125
63
32
16
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Flame Cutting-Machine
Rough Turning
Contour Sawing
Rough Grinding
Shaping and Planing
Cold Sawing
Drilling
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Aluminum
Brass
Cast iron
Soft steel
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Full range commercially used
Average Machine Finishes
Noncritical Parts
Average Parts
Critical Parts
Relative Cost (in units of time required)
Applicable Typical Dimensional Unilateral Tolerances (in.)
Small sizes and dimensions
Tolerance range
Large sizes and dimensions
General trend of cost curve
0.00001
0.00002
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0.00008
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0.250
Surface Finish (microinches, rms)

the specification of the most suitable processing method and its natural tolerances or size reproducibility, in any system the limits must likewise be subject to the production methods used. The first consideration, therefore, would be the processes available and their practical limits. And these are *never* subject to any formula—they are the result of statistical study of the natural tolerances produced under everyday conditions—and here the nominal size may have relatively little effect on the variation. More important, surface roughness must be a vital part of the picture. The accompanying chart gives a general idea of the interrelationship of these factors.

It appears that the practical approach would be to devise a standard system of fits—which *are* subject to arbitrary decision and machine functioning—based on the unilateral system. On such a system the practical tolerances—and processes needed to achieve them—could be applied as dictated by machine requirements and economics.

By all means the system should be simple. All possible data should be in direct table and chart form for rapid, direct use. With such a system, adaptation would become natural—designers and engineers continually are seeking improved and dollar-saving roads to better design.

Advantages of a Uniform System

By Carl H. Ringe
Assistant Director of Research
R. Hoe & Co. Inc.
New York, N. Y.

A PLURALITY of individual systems of fits, tolerances and allowances has been developed over the years by various industries and groups of industries for their individual purposes. Based on the experiences with these systems, an early American Standard for tolerances and allowances was established approximately 30 years ago, but was never adopted unanimously by industry. This system uses for each class of fits a complete set of bilateral tolerances for the internal dimension, i.e., the hole, and for the external dimension, i.e., the shaft. It is basically our present system.

Meanwhile, important industries of some other highly industrialized countries had developed another, basically different system, resulting in the ISA tolerance system which was mostly influenced by the ball and roller bearing industry and has proven to be highly practical and efficient.

Under the influence of the exigencies of the war and the experiences gained by the mass production of war materiel, the new and improved ABC system of tolerances and allowances has been developed. This new system comes close to the ISA Standards which have been adopted internationally by the industries of many other countries. The basic difference between present standards and the new system is that the latter is unilateral, which gives only the choice between basic hole and basic shaft.

This means in the first place a clear definition. In the first case, the hole (or internal dimension) always has the tolerance in the same direction from the design size, while in the second case the shaft (or external dimension) is the basic part. The tolerance given to the dimension of the corresponding part gives the desired allowance. The advantages of the new system are obvious. The clear and unmistakable definition makes it easier to understand and it simplifies all stages of production.

Designing is simplified because one variable is eliminated. The designer can visualize more easy the chosen allowance, due to the uniformity of the method of applying the tolerance. This results not only in economy of time in preparing correct detail drawings but also in avoiding a source of faulty dimensioning. Tooling is less complicated because one detail part is always given the same tolerance in the same direction from the design size for a given size, independent of the final fit between the two mating parts.

This means economy in the toolroom because of a reduction of the amount and variety of drills, reamers, bushings and gages. It means an easier production of machine components without departing from high precision. In connection with the establishment of a table of preferred basic sizes for the most used range of diameters, a complete set of go and no-go gages and plugs can be used for production instead of the conventional micrometer. This helps greatly to eliminate human errors and facilitates precision production. Assembly of individual detail parts is improved because of the uniformity of the system.

Since the entire system of unilateral tolerances comprises a wide variety of grades of fits within the three categories—clearance fit, transition fit and interference fit—it is obvious that it can be applied flexibly to all conditions of manufacturing. Due to the methodical establishment of its sizes, various industries can better co-ordinate their grades of tolerances. This is important because few manufacturers produce all detail machine parts. The great number of subassemblies manufactured by specialized industries must assemble precisely without further production operations.

The problem of spare parts for repair and replacement becomes simplified. Once the fundamental choice between basic hole or basic shaft has been made, a spare part can be supplied with less difficulty regarding its fitting in the place of the original one.

It becomes a matter of vital importance in times of political stress where a defense program needs the production capacity of all industries of the nation. In such case a clearly defined part of

the entire system, unmistakably understood by all involved, can be readily determined for the production of all individual detail parts which can be assembled to complete units without trouble and with constant precision no matter in what part of the country and by whom these detail parts had been manufactured.

Basic Shaft System Needed

By E. N. Stoner and J. A. Gnandt
Standards Engineers
Harris-Seybold Co.
Cleveland, Ohio

IN the opinion of the standards engineers of the Harris-Seybold Co. the ABC system of limits and fits as proposed by the ASA could be used as a base on which to formulate a company standard. However, there are certain reservations which temper this acceptance. Before presenting the factors considered in arriving at our viewpoint, a brief picture of the company's products and operations is necessary.

Harris-Seybold manufactures printing presses, paper cutters, and allied equipment for the graphic arts industry. Designs are originated and detailed in any one of several separate company engineering sections for manufacture in one or more of its divisions located in different parts of the country. Each product is a complex precision machine and is produced in a number of models or sizes. Only a relatively small number of machines of any given model are built at a time. This means that machine tools are of a general-purpose nature, and tooling and gages are kept to a minimum of variations in order to have flexible facilities.

Need for a Common Limit and Fit System: Over the years an adequate system of limits and fits, suited to a line of products, has been developed by each division of the company. However with the recent transfer of some production from one division to the other, the need for a common system has become evident. Consequently the standards engineers of the company considered devising such a system, but after careful consideration abandoned the project in the hope of finding a widely used system which would be adaptable to the requirements of all the company's products.

Merits of the ABC System: To date the ABC system fulfills the limit and fit needs of the company more nearly than any other system from several aspects. The system accommodates the use of preferred basic sizes; diameter steps in the ASA fundamental tolerance table have been selected so as not to match common inch sizes. Its tolerance grades are reasonable and proper as diameters increase. The many combinations of limits and fits possible with the ABC system extend, in scope and detail, beyond the requirements of the company's products. The favoring of a national system over local or restricted standards would help simplify some of the problems encountered when parts are purchased or subcontracted. Furthermore, national usage of the ABC system can reveal to an initiated person the type of fit desired for an assembly by showing the system's designation on component drawings; for example, H7-f7 would indicate a precision running fit.

Use of Limits to Ten-thousandths: Having noted the merits of the purpose and basic concepts of the ABC system, we then considered the values of the limits in the ASA proposal for the grades and diameters shown. These have been calculated from an empirical—but practical—formula and have been rounded off to ten-thousandths for values up to 0.0045 inch and to thousandths for values above 0.0045 inch.

We feel that indicating limits to ten-thousandths for general applications is unnatural. Unless the design requires otherwise, designers usually specify limits in thousandths. Published tables have indicated that the accuracy of such common processes as turning, milling, etc., is in terms of thousandths. Many items like finished rounds, bushings, etc. have their limits specified in thousandths. Thus in practice, we believe, there will be a tendency to modify the ten-thousandths limits in the ASA tables to thousandths. This tendency probably will be more likely for parts made and inspected with general-purpose equipment. With the intention of standardizing the practice of rounding off low limits to thousandths, we would like to see such data, at least, recommended in a supplementary table in the ASA standard.

Basic Hole vs. Basic Shaft: In appraising the ABC system further we next considered the basis of recommended fits. The basic hole system is used in preference to basic shaft in TABLES I through IV; finding the nearest equivalents per basic shaft in the appendix is possible but cumbersome. We realize that basic hole makes possible some tool simplification to a number of concerns. However, we feel that the standard should, and could, give equal treatment to tables for basic shaft. It is our understanding that the group of concerns involved with the use of commercial finished shafting constitutes approximately 20 per cent of industry having limit and fit applications.

Harris-Seybold is a particular concern that illustrates the need for a series of recommended fits based on basic shafts. Its printing presses and paper cutters require the use of a large amount of shafting in and between frames. This shafting is purchased from mill sources in varying degrees of tolerances and associated surface finishes. In some cases the shafting is used directly without modification, especially where several components requiring varying amounts of clearance or interference are fitted on it; here the company

saves time in not machining long lengths. Fits are then made as basic shaft. In other cases shafting is machined for functional reasons to provide steps, shoulders, and other necessary alterations. In these cases, fits are basic hole as they also are when mating parts are purchased. Thus, it can be seen that due to the nature of the company's products both basic shaft and basic hole are utilized. Therefore, to make more generally feasible the adoption of the ABC system to our company, it is hoped that the ASA will publish a series of recommended fit tables based on basic shaft as currently shown for basic hole.

A Company Limit and Fit Guide: In looking over the tables of recommended fits in the ASA draft proposal we felt the need for a guide that would tie in particular fits to specific applications. This, we realize, cannot be done on a national scale because of the great mass of detailed data that would be required for the many applications possible. A guide would have to be prepared on company level (or at most on an industry level for such items as bearings, bushings, etc.). This, in some cases, may involve a fresh study to determine actual limit and fit requirements since designers are sometimes prone to specify limits and fits to closer values than may be necessary and shops tend to modify or interpret those limits and fits specified on drawings which they deem unreasonable or unimportant.

Comparison of Limits

Shaft			Hole			Clearance	
Company	ABC		Company	ABC		Company	ABC
.750 /.748	.7500/.748	h9	.751/.752	.7508/.7520	F8	.001 /.004	.0008/.004
.7505/.7495	.7506/.7494	j8	.751/.752	.7508/.7520	F8	.0005/.0025	.0002/.0026
.7495/.7490	.7497/.7492	g6	.751/.750	.7500/.7512	H8	.0005/.0020	.0003/.0020

Note: The basic shaft concept is used. Reasons for the above clearances have not been given here for sake of brevity.

Compiling a guide would require listing many factors. Fits may be affected by such factors as material, surface finish, length of engagement, load, speed, etc. Limits may be affected by such factors as production processes available, quality conscientiousness of the shop, inspection equipment, etc. In any case, the guide should not emphasize limits and fits by their extreme values but rather stress the probability of normal or mean values.

ABC Fits vs. Company Fits: In selecting shaft and hole combinations from the ABC proposal that were the closest to those in current use, we observed that the fits, when compared, were at variance by several ten-thousandths. The accompanying table shows this in the 0.750 inch size for different clearance requirements.

In some cases the variations of the fits from company practice will not be such as to prevent adoption of the ABC system. However, in others adoption will have to be delayed until a new model is initiated because of the effect on jigging and tooling.

Harris-Seybold services parts over a long period due to the class of its products. Similarly some parts are used interchangeably in several models. Without considering gaging conversion the preceding reasons alone indicate that the company will not be utilizing the ABC system fully for a number of years.

Summary: In summary, our overall opinion regarding the ABC system both in relation to our own company and the national interest is one of general endorsement with a few reservations (which may be cleared up later by the ASA). Reasons for adopting the ABC system are: (1) Based on practical fundamentals—diameter steps, tolerance gradients, etc.; (2) has a comprehensive range of limits and fits; (3) enables engineers to specify fits in the most economical shaft-hole limit combination and still achieve the proper fit; (4) establishes a practical code for indicating the fit desired; and (5) points the way to tooling and gaging simplification. Objections to the ASA Proposal can be summarized as: (1) Does not have basic shaft tolerance and fit data available in convenient for for quick reference by concerns using commercial finished shafting; (2) has low limits in ten-thousandths which are not necessary for general applications and which contribute to gage change costs in excess of a practical minimum which must be anticipated.

In conclusion, we find that Harris-Seybold could adopt the proposed system by expanding tables to include limits and fits data based on basic shafts, by rounding off limits from odd ten-thousandths to even thousandths or half thousandths, and by introducing the system over a long term period. For the sake of standardizing as much as possible, we would like to see recommendations on rounding off limits to thousandths and for fits based on basic shafts presented in this standard.

Standard Would Aid Procurement

By George Pascoe
Mgr., Design and Standards Dept.
Manufacturing Engineering Office
Ford Motor Co.
Detroit, Mich.

NEED of a system of limits and fits applicable to manufacturing engineering work has been recognized in our company for quite some time.

The establishment of such a system would be of definite value to our designers of machines, tools, fixtures, gages and other equipment required for high production of automotive and allied products. It would be likewise valuable to quality control personnel responsible for the acceptance of tools procured from outside sources. Available tables of fits at present are few, incomplete, and not always sufficiently clear-cut.

The lack of a nationally recognized system of limits and fits is often a source of guessing, misunderstanding, delays, and, in some cases, arbitrary decisions regarding dimensions affecting the accuracy of details and their relation in the assembly of the various types of tooling for high production.

The ability on the part of a designer to specify

SKF

SELECTION OF SHAFT TOLERANCES

(For numerical value, refer to page 70)

Shafts for Radial Bearings

Conditions		Examples	Shaft diameter in mm. Ball Bearings	Shaft diameter in mm. Cylindrical Roller Bearings	Shaft diameter in mm. Spherical Roller Bearings	Tolerance symbol according to ISA	Remarks
Bearings with Cylindrical Bores							
Fixed inner ring load	The inner ring to be easily displaced on the shaft	Wheel on stationary shaft	All diameters			g6	
Fixed inner ring load	The inner ring does not need to be easily displaced on the shaft	Tension pulleys and rope sheaves	All diameters			h6	
Rotating inner ring load or indeterminate load direction	Light and variable loads	Electrical apparatus. Machine tools. Pumps. Ventilators. Industrial trucks	≦ 18	–	–	h5	By light and variable loads are meant those which, as a rule, do not exceed 6 to 7% of the specific capacity C. In very accurate applications j5, k5 and m5 are used instead of j6, k6 and m6 respectively.
			18 . . . 100	≦ 40	≦ 40	j6	
			> 100	40 . . . 160	40 . . . 100	k6	
			–	160 . . . 225	100 . . . 225	m6	
	Normal and heavy loads	Applications in general. Electrical motors. Turbines. Pumps. Combustion engines. Gear transmissions. Woodworking machines	≦ 18	–	–	j5	
			18 . . . 100	≦ 40	≦ 40	k5	
			> 100	40 . . . 160	40 . . . 100	m5	
			–	160 . . . 225	100 . . . 200	n6	
			–	225 . . . 400	200 . . . 355	p6	
			–	–	355 . . . 500	r6	
			–	–	> 500	r7	
	Very heavy loads and shock loads	Journal boxes for locomotives and other heavy rail vehicles. Traction motors	–	60 . . . 160	60 . . . 100	n6	Bearings with greater clearance than normal must be used.
			–	160 . . . 225	100 . . . 200	p6	
			–	–	200 . . . 355	r6	
			–	–	355 . . . 500	r7	
Pure thrust load		Applications of all kinds	All diameters			j6	

Shafts for Thrust Bearings

Conditions		Shaft diameter in mm.	Tolerance symbol according to ISA
		All diameters	j6
Pure thrust load		All diameters	j6
Combined load on spherical roller thrust bearings	*Fixed load on inner ring*	≦ 200	k6
	Rotating load on inner ring or indeterminate load direction	200 . . . 400	m6
		> 400	n6

appropriate fits and limits in any particular field is developed through experience only. However, because of the numerous factors involved, proper performance of mating parts is often obtained only after costly dismantling and reworking. It is conceivable that proportionally few designers possess the necessary experience to establish optimum limits in all the fields encountered in manufacturing engineering work. Designers of limited experience are obviously at a further disadvantage, and it is to be expected that they will tend to circumvent this difficulty.

Specific Data Needed: It is common, contrary to recommended practice, to find on manufacturing engineering drawings such specifications as slip fit, press fit, or similar description of the fit desired, instead of specific limits. In these cases the choice of sizes is, in fact, left to the discretion of the toolmaker, whose judgment depends on his own experience, and who may not be sufficiently acquainted with the requirements of the particular application for which the fit is intended. This practice will often result in the necessity of dismantling and reworking the mating members to produce the fit required for proper functioning, with the inherent expense and loss of time. Incidentally, the experience so gained is usually confined to the personnel in immediate contact with the job developed, and not made available to other designers in the form of recommended dimensions for similar applications.

Also, many standard and commercial parts normally procured for high production, such as tools, tool adaptors and holders, fixtures and gage components, drill spindles and similar components, are often specified by the suppliers by nominal sizes only, with the added disadvantage that different suppliers of similar items may apply different limits, according to their individual concept of the requirements. This practice creates constant difficulties for receiving inspection personnel; investigation of particular applications is often necessary to determine the permissible variations from the nominal size that would allow adequate performance of the parts procured.

Similar disadvantages exist in numerous other fields, and we feel that the amount of reworks, fitting, and tryout could be reduced and, in some cases, eliminated, if the designer were enabled to specify appropriate limits from complete charts.

System is Established: One instance where fits are specified is in the field of antifriction bearings applications. Specific limits for the size of the shaft and housing for each bearing have been established by the bearing manufacturers, and are in accordance with such systems as ISA (International Federation of the National Standardizing Association), ABEC (Annular Bearing Engineers Committee), and other pertinent standards. These limits are used throughout industry with the result that the required fits for correct performance are secured, even in the event that different members should be designed and made by different sources. Several bearing manufacturers have further established recommendations regarding the type of fits to be selected, according to the application intended for a particular bearing (see chart). It is evident that if this system were not available, the limits for shaft and housing would be specified according to the concept of the individual designer for each application. It would then be reasonable to expect that in many instances faulty performance of bearings would result, because of improper limits.

In our opinion, the system of fits and limits in the field of antifriction bearings application, in contrast to the disadvantages in many other fields because of the lack of an adequate system, offers an excellent example of the benefits that could be derived from the adoption of a similar system to those fields. We therefore advocate that standard tables of recommended fits and limits be adopted, for industry-wide use. These tables would reflect the experience gained by engineers and designers relative to limits and fits pertaining to various applications that would ordinarily be difficult to express in writing, thereby facilitating the training of designing and engineering personnel, among other advantages.

System Attributes: Concerning general limits and fits, we believe that this system is a practical approach toward the establishment of a system of limits and fits that could be adopted by the American industry. This system represents the summary of the result of extended studies in this field, including other standards such as ISA and previous ASA publications. The system includes, among other factors, the following advantageous features: (1) Tables presented illustrate limits applicable to many fields; (2) classes of fits are clearly defined and classified; (3) certain fits of general usage have further been designated on a functional basis; (4) selection of fits appears adequate to cover most engineering requirements; and (5) fits for a particular application can be developed from the general data to suit particular requirements, and still be within a graded system of fundamental tolerances.

It is conceivable that further advantages could be gained if additional designation of fits on a functional basis were developed. For instance, we feel that additional running clearance fits consisting of different allowances and grades could be established, to suit the requirements of the various types of applications, such as fixtures, machinery, gaging apparatus and other tooling required for high-production equipment. However, such tables may be compiled by suitable combination or pertinent modification of the proposed ABC tables. Other types of fits can similarly be developed to suit the needs in any other particular field.

It would also be desirable that reference be made, on the proposed ABC system, to gagemakers' tolerances and other systems of tolerances, limits, and

fits now in general use. We agree, however, that the proposed ABC system in its present form offers sufficient advantages for immediate adoption by the American industry, with pertinent amendments and amplifications to follow in the future.

Fits Can be Simplified

By Alfred Beck
Chief Checker
Hautau Engineering Co.
Ferndale, Mich.

THE ABC System of Limits and Fits is an excellent presentation of the classification of tolerances and fits. It gives the engineer a language with which to express a desired fit without special work. The junior engineer can specify the proper tolerances because the way is clearly shown. Industry can inspect with standardized gages, the tolerances of which have long been agreed upon.

The only fault which the system has is the same fault which all such systems have in common. The average engineer does not know which fit to pick for his application and that, of course, is equivalent to having a bucket full of beautiful paint for your living room—just the right thing—and no brush.

In my experience in tool design and machine design I find that it is sometimes necessary to point out the rules for a standard hole. It is poor practice to specify holes with tolerances less than standard, that is, from a standard hole to minus. The excuse for such a procedure is that the shop has old reamers to use up. Let the shop use its own judgment when and how to salvage used tools. The possibility that a good tool may be honed down to meet the specifications of the drawing forbids such a practice.

In tool design, due to the variety of products a variety of fits must be used and for many tool designers any such system seems too complex. Therefore, it is often left to the shop to decide what the limits should be. The procedure followed in such cases is to call for press fit, push fit, slip fit, etc. As soon as several tools, gages or fixtures of the same kind are needed it becomes obvious that it would be better to call for definite tolerances than calling for press fit or slip fit, etc. Not only the fact that selective assembly is an undesirable method for the shop, but even more so the replacement of perishable parts demands specific tolerances. For this reason the manufacturers of ball bearings, bushings, etc., give definite tolerances for use with their products. This protects the seller from unjust claims and also protects the user from failure due to faulty installation.

Almost every jig or fixture needs some accuracy in part of its construction. In order to obtain the most accuracy for the least expenditure, the old fear of metal-to-metal fit should be abandoned. No one would accept a metal-to-metal fit for a press fit. Let us consider a one-inch plug to fit into a one-inch hole. The hole should be 1.0000 inch +.0005 or −.0000 and the plug should be 1.0000 inch +.0000 or −.0005 for a reasonably loose fit. Should the plug be made .9995 inch +.0000 or −.0005, the accuracy of location would suffer and the expenditure for obtaining the fit would be the same. Statistics tell us that it is a better than 100:1 bet that the two will fit without trouble. Furthermore the method of measuring will probably raise the odds considerably.

Ordinarily a one-inch hole is considered not less than one-inch only when a one-inch plug enters by its own weight and vice versa. A one-inch shaft is considered to be not more than one-inch only if a one-inch ring slips over it by its own weight.

Should the above mentioned plug be used in a gage either as a location or as a turning member the tolerances should be +.0002 and −.0000 and +.0000 and −.0002 respectively. The absence of lubricant, which is a prerequisite on gages, makes such a fit quite loose. When it comes to press fit, the majority of cases is well served by a hole +.0005 and −.0000 and a shaft +.0015 and +.0010.

Give Designers Sample Gages

By Charles Eumurian
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Engineering Research & Development Dept.
General Mills Inc.
Minneapolis, Minn.

THE question of tolerances is one that is of utmost importance to those who are in the machine design field. It is also a question surrounded with a considerable amount of confusion with most tolerances for dimensions being selected by the design engineer rather arbitrarily from his experience in that field. A national standard should clear the confusion, provided the design engineers are willing to conform to the standard. There are enough different classes of fits so that there should be very few exceptional cases of special tolerances.

Advantages of such a standard can be summed up in a few paragraphs.

1. Reduce engineering time required to determine what tolerances are to be used. Once the class of fit is determined, the actual dimensions would be obtained from the standard tables.

2. Reduce drafting time. A draftsman working from a layout drawing assigns tolerances to dimensions which are checked and (most generally) changed by the engineer in charge. With a standard, the class of fit would be assigned to each portion of the layout, and detail dimensions and tolerances would come automatically.

3. Provide a simpler method of referring to a class of fit.

4. Insure the mating of interchangeable parts from different manufacturers. The class of fit would be part of the item specification.

In order to assure the use of a national standard for tolerances and fits, it would be wise to fabricate a set of gages (for an arbitrary diameter) with a corresponding set of holes in a plate. There would be one gage with minimum and maximum diameters and one hole of maximum diameter for each class of fit. A reference basic hole would be available for use with all gages. This system would apply only to the running clearance fits.

Engineers being the practical type of people that they are, will more likely accept a thing if they can see and feel it. By giving them a sense of feel of the various classes of fits it will be easier to determine the type of fit they want and will specify it in terms of the national standard. Such a system of gages might be expensive, but I feel that it would be a worthwhile addition to any engineering department.

Fit Conditions Need Study

By C. J. Larson
Director of Standards
International Business Machines Corp.
New York, N. Y.

WE HAVE not adopted a formal system of limits and fits, but the need for some workable standard on this subject is becoming increasingly apparent. A decision as to which system should be adopted must be reserved, pending a complete evaluation of the advantages and disadvantages of present and proposed systems. We have become convinced, however, that a system of limits and fits is a primary problem, of which dimensioning is merely one phase.

Formal standardization on the basic shaft system of dimensioning has been established since June 1946, when there was difficulty in obtaining a range of sizes in ground and polished shafting. Recent investigation indicates that the known deviations from this system have increased to a point where a re-evaluation of our position is necessary. Such a re-evaluation has been in process for several months, but no definite conclusions have as yet been reached.

Discussions with operating and engineering executives within IBM revealed definite and varied opinions on the choice of basic hole vs. basic shaft systems of dimensioning. However, a definite lack of factual information on the effect of each system was recognized. A study is now in progress at IBM plants by the respective manufacturing standards groups to develop such facts. Some of the factors

being considered in this study are: (1) Standardization of tool sizes and improved availability; (2) standardization of plug gage sizes, and consequent reduction in cost to IBM; (3) standardization of raw stock requirements; (4) reduced inventories of stock, tools, and gages; (5) more effective use of component parts standardization; (6) more efficient use of manpower used to select standards; (7) systematic application of the most logical manufacturing tolerances.

When the benefits to be gained from the selection of one system or another, or a workable combination of both systems, have been established, a logical decision can be made by management. From discussion with executives of other companies, it appears that the same general conditions exist, i.e., no definite economic evaluation and widespread deviations from the recognized standard system.

A preliminary review of the new ABC proposal of the ASA has been made, but no definite commitments are possible at this time. The ABC system is very broad in its scope and more than covers our needs in its overall range. Actually, if this system or a variation thereof is adopted, the table to be incorporated into IBM Standards will be a very small portion of the proposed comprehensive table.

It is also very doubtful that the ABC system can be literally adopted in its existing form. For example, our standard tolerance on shafting is plus 0.0000, minus 0.0005-inch. The ABC system applies these tolerances to only two size ranges in GR 6 and 7 tolerances. It is also doubtful that we would require as fine a breakdown by diameters.

Considerable opposition to the symbolism of the system has appeared from our preliminary investigation and the need for such a system at this time has been questioned. On the other hand, a system must be specific and comprehensive to be effective —the ABC system has both of these virtues. The nomenclature terms such as running fit, clearance fit, snug fit, etc., are merely opinions subject to local interpretation, but an "H7 e7" fit is far more specific. Tables based on the nomenclature of a fit are also subject to great bulk and detail if they are to be comprehensive and include, not only the fit desired, but the nature of the manufacturing applied to the parts. The ABC system provides a simple method for relating nomenclature to symbolism. Too many cases are evident where fits are selected by guess or opinion, and there is a great need in this country for a comprehensive study of fit conditions.

Investigation into the details of developing standards on small mating components such as collar, shafts, sleeve bearings, etc., indicates that some such system is badly needed. While it may not benefit engineering directly, manufacturing appears to be paying the bill for lack of a definite system of limits and fits.

The ability of the ABC proposal to translate values appears to be valuable to IBM in view of our extensive world-wide operations through our World

Trade Corporation. The interpretation of information presented in the ABC proposal nomenclature is specific and will reduce misunderstandings.

Whether or not the ABC system will be adopted by IBM is not certain at this point. It does seem probable that some system of limits and fits wil' be adopted, and the ABC system provides a gooc basis for exploration.

What About Metric Compatibility?

By E. W. Drescher
Director of Quality
Hamilton Watch Co.
Lancaster, Pa.

CONSIDERATION of the proposed ABC standard from the viewpoint of our field of manufacture might prove quite helpful in appraising the proposed standard in the very small range of nominal sizes. Exploration of several applications of the proposal to our manufacturing reveals several problems regarding suitability of the present standard.

For example, in one instance our shaft size is stock size, and the limits resulting from the method of tolerancing insisted upon by the supplier's industry do not coincide with the limits the proposal gives unless the rounded value for the nominal stock size is abandoned. These, in turn, influence the limits for the hole size and do not permit the use of a rounded basic hole size.

In other trial runs various other obstacles appear. An odd type of situation which is very commonplace with us, but which the proposed standard apparently does not provide for, is the combination of one component from a source where the metric system prevails with another produced under the inch system.

From a brief perusal of the issue I see little to attract our industry to the proposal in its present form.

Fit Recommendations Desirable

By Harold E. Hering
Standards Engineer
Miehle Printing Press & Mfg. Co.
Chicago, Ill.

THE subject of tolerances and fits has been a problem to many companies engaged in manufacturing. The problem becomes more acute as the precision required of the final product is increased. The range of allowable tolerances becomes so finite that only the latest developments in the field of masuring instruments can be used for inspection purposes.

Our company is no exception to the rule and has had many difficulties in trying to determine workable fits for the various mechanical assemblies that have been designed. Although selective assembly is an answer to some of the problems that are encountered, this is not the ultimate goal that is strived for in manufacture. The final dimensions arrived at in some of the cases are the results of modifications of the original dimensions assigned to the parts. Although many companies have very elaborate standards developed for the various classes of fits, problems are still encountered for unusual conditions of design.

Growth of the standard on tolerances and fits at our company has not followed any specific recommendations by the ASA in the past, but the present standard has been partly based on some of their recommendations. The unilateral tolerance system has been adopted in order to simplify the dimensioning of parts and to obtain the tolerance of the fit readily.

In addition to adopting the unilateral system of tolerance specification, the company has also adopted the basic hole system as also suggested in the ASA B4.1—1947 standard on limits and fits. Using the basic hole size, the proper fit desired can be obtained by varying the size of the shaft or male member. We have selected seven different types of fits; loose, running, sliding, push, drive, press, and force. Although provision should be made for variation in the total allowable tolerance, for various lengths, this has not been taken into account in our standard. Table 1, page 6, of ASA—B4.1—1947 lists preferred basic sizes. These sizes were useful in the formation of our standards on fits. The basic sizes along with Table 2 tolerances and allowances were adhered to as closely as possible in the formation of our standard. It was a guide and supplied very useful information in the compilation of our standard.

Although we do have a company standard on limits and fits, variations are still made from these recommended figures in order to arrive at the final dimensions placed on the drawing. In addition. further modifications might have to be made in the event difficulties are encountered in production or final assembly of the mechanism. These modifications in most instances cause a departure from the standards that are used by the company. The standard released by the ASA in 1947 titled "Limits and Fits" ASA B4.1—1947 Part I was a decided step in the right direction. The standard was brief and concise and did not attempt to cover too much in its scope. The basic definitions were given and the listing of the preferred sizes as well as the tolerances and allowances acted as a guide to a manufacturer setting up a standard to be used as a reference for engineering personnel.

The proposed standard for an ABC System of Limits and Fits B4/30 is a very noteworthy step

with an aim toward unifying a proposed standard to be adopted by all three countries. Main objections to the proposed standard may or may not be those voiced by engineers in general. The standard does not direct an individual to a specific class of fit for a particular design. The standard should include some tabulation of the recommended classes of fits for the various types of mechanisms. It does not have a listing of the preferred basic sizes. The listing of preferred basic sizes will tend to minimize the number of basic sizes that will be used in industry. This will be of inestimable value as it would result in the increased usage of fewer sizes thereby benefiting a large portion of industry.

The standard is too broad in its scope. It tries to incorporate too much into one standard. It would be more useful to industry as a whole if the standard were subdivided into a few major divisions and a separate standard released on each particular phase of the general standard. In general, a large portion of the groundwork has been laid in the proposed standard and can be used as a guide for subsequent standards and revisions in this field.

Formula Needs Explanation

By H. F. Irving
Head, Engineering Services Dept.
Baker Perkins Inc.
Saginaw, Mich.

THIS appears to be a very complete system covering almost every conceivable class of work. Its universal adoption should make it of great value to industry in transferring information from one organization to another as to the tolerance and fits used on equipment supplied.

On first reading over this proposal I found the system of letters and numbers very confusing and not until I had given it considerable study did I get it clearly in mind. If other people have this same trouble it may seriously retard its acceptance. I believe that more space in the introduction explaining the use of the letters and figures might be of considerable value in helping the individual grasp the system as a whole. The proposal is far too broad for general use by individual designers and draftsmen and, as suggested, each organization must select two or three grades of fits which satisfy its needs.

In Appendix II the method of arriving at the various shaft and hole diameters is given but there is no reason given as to why the method was selected. In other words, why is the expression $i = 0.052 \sqrt[3]{D} + 0.001D$ selected to represent the fundamental tolerance? Is there some basic reason for selecting this expression or was the expression selected as one which would closely parallel present practice?

For publication in this country I believe the decimal point should be placed in the customary place at the lower edge of the figure instead of in the middle where it might be confused with the algebraic use of the dot as a multiplication symbol.

This proposal certainly offers an adequate selection of limits and fits for all of our work here at Baker Perkins. It appears, however, that only when industry accepts this proposal and then takes the next step of agreeing on a selection of certain fits for certain classes of work, are real benefits to be derived.

Too Many Hole Standards

By George E. Hieber
Assistant Chief Engineer
Landis Machine Co.
Waynesboro, Pa.

LANDIS Machine Co., builders of machine tools, is now using a standard hole system based on ASA tolerances. We use two series of holes, a high-class series which agrees closely with the proposed H6 series and a low-class series which agrees closely with the proposed H8 series.

Our high-class series does not have as close a limit as the H6 series on the sizes below ¼-inch diameter as we feel that a reamed hole in these small sizes is very difficult to make to a tolerance of standard to plus 0.00025-inch. Our limit on a 3/32 diameter hole is 0.09375 to 0.09415, and that is the smallest hole we use. Our high-class series includes all millimeter ball bearing housing bores and the limits agree with those suggested by the ball bearing manufacturers. As an example, a 209 ball bearing requires a housing bore of 3.3463 to 3.3473 as per old ABEC Standards and will provide a 0.0002 interference to 0.0016 loose fit.

Since we have adopted these sizes, ball bearing limits have been changed to require a bore of 3.3465 to 3.3474 as per the new ABEC Standards giving a fit of metal to metal to 0.0015 loose. This clearance is due to the fact that the bearing manufacturers have tightened up on their own limits on the OD of the bearing. All our plug gages are of the old standard and we have to stick to it for cost reasons. Luckily, the variation is small and the new bearings work just as well if not better with the old bore sizes.

The proposed ABC Standard would require a hole of from 3.3465 to 3.3474 and it would fall exactly into the requirements of the new ABEC Standards. Any commercial component that requires a good fit of either a clearance or interference type would work in the same manner, since

there is only a very slight difference between the proposed ABC and the ASA Standards in use at the present time.

Of the new ABC Standards the only comment that our company can feel is in order is that too many hole standards are proposed. We can see no use for H7, H9, H10, H11, H12 and H13. There might be a use for H5 but it is only slightly more accurate than H6 and for ball bearing fits, holes where bronze bushings are press fitted, running fits for plain bearings, etc., it is not needed. It might be that the H5 is proposed for hydraulic valve bores but for such use it is not good enough as for example, the 2.000 diameter hole has a limit of 2.0000 to 2.0005. That is too much for a valve body fit even if the male part is held to extremely close tolerances. For work of that kind a total clearance between the male and female parts of 0.0002 to 0.0004 is about all that can be tolerated at pressures of 1000 psi and up. The H4 series might be more useful.

Most holes of any accuracy are finished by reaming or grinding or broaching. They all lend themselves to the economic reproduction of size within the Class H8 series. No great money savings can be achieved by loosening the tolerances to the H9 or H10 series. As an example, let us consider a 2-inch hole in the proposed series. It can be economically held to 2.000 to 2.0018 by very crude reaming, grinding or broaching practice. No possible time could be saved or tool cost lowered by making those limits 2.000 to 2.0030 for Class H9 or from 2.000 to 2.0045 for Class H10. The question arises as to whether the reaming could be eliminated if the coarser limits of the H9 or H10 series were used instead of the H8 series and, of course, the answer is no, as average drilling practice cannot hold a 2-inch hole to 2.000 to 2.0045. So far as cost saving is concerned there is, in our minds, no economic reason for the existence of the H9 and H10 series.

Is there a use reason for their existence? Again, in the machine tool business, we might have a shifter shaft with a clearance of 0.0016 to 0.004 on a 2-inch diameter, as is accomplished with the H8—C8 shaft fit. We can see no use for a clearance of 0.005 to 0.0115 on the same 2-inch shaft as is accomplished by the H10—C9 shaft fit. It is not good enough for even the crudest of running or sliding fits and if it is only to insure clearance for a fastening member or some such use, why hold even to the limits shown? Why not drill the hole 1/64 oversize? This would cause it to fall in the proposed H13 series. The mere fact that a 1/64 oversize drill is used would insure that any grade male member would enter the hole. Would gaging be resorted to? Are the standards for the purchase of gages? How would a 2-inch hole of the H11 series be produced? It calls for a limit of 2.000 to 2.007. Would a bastard drill or reamer be used? How about the cost of actually producing these series of coarser limits in regard to tooling? None of the limits shown agree with any commercially available drills or reamers.

In our own business and considering all factors, we feel that we only need the H6 and H8 series of holes and a few possible applications of the H4 series. Any of the coarser series have no practical application.

Now, in regard to the shaft series, all male members are produced and measured by means capable of securing any diameter without increasing tooling cost. On the other hand, hole production requires in general fixed sizes of tools and gages. Why attempt to set up a series of male diameters? When grinding a shaft to size and measuring it with a micrometer it makes no difference in tool cost if the limit is 1.999/1.998 or 1.999/1.990. The only thing affected is production time. We suggest to our engineering department a series of limits for a good running fit, a loose running fit, a light drive fit, a press fit, etc. In no case are the limits the final word nor do they have to be religiously followed.

If the running fit for example can stand an extra tenth or two clearance, the larger tolerance is shown on the detail drawing. This is an economic advantage both in manufacturing and assembly. A press fit with a great length in relation to its diameter will be given a smaller metal interference than one of a short length. What advantage has a series of sizes for shafts as suggested by the ABC Standards over the idea of making the shaft anything you please to suit the individual case?

Finally, we have too large an investment in tools and drawings to change over from what we have to the ABC series at once. We might, as the tools wear out, replace them with the new standards and all new drawings could be made to the new standard. Parts made to the new standard for repairs and replacement would interchange perfectly as the sizes we use now are only different by a tenth of a thousandth in most cases than those proposed. We have no objection, therefore, if the new standards are adopted.

Basic Hole System Used

By K. B. Kaiser
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Rockford, Ill.

COMMENTING on the proposed new ABC system of Limits and Fits, we believe the fundamental idea of establishing a system of standardized fits would be helpful, especially to smaller companies which do not have a large enough or well qualified enough staff to work out a system of fits suitable to their own needs.

In our own case, in the transition from the "old

days" when most drawings were dimensioned in fractions or parts were made "to suit," it was necessary for us to build our own set of engineering standards which we now use. These are considerably simpler than the proposed ASA standards in that we use only one basic set of hole limits and obtain our various fits by varying the shaft size. We have eight different classifications for shaft size depending upon the application. These are considerably simpler than the proposed ASA standards in that we use only one basic set of hole limits and obtain our various fits by varying the shaft size. We have eight different classifications for shaft size depending upon the application. These standards are used for the majority of our work with the exception of special cases where, because of extreme precision, we close-in on the hole limits. Another exception is in our antifriction bearing fits where we follow the recommendations of the bearing manufacturers quite closely.

We realize that in order to apply to all industries, the proposed standards would have to be considerably more comprehensive and that any one industry, such as ours, would probably use only some small section of the standard.

In its present form, we have the impression that the proposed standard is too complicated to be understood and properly used by the average draftsman. It may be possible, in future revisions, to use simpler and more descriptive terms and illustrate them with examples.

CONTEMPORARY DESIGN

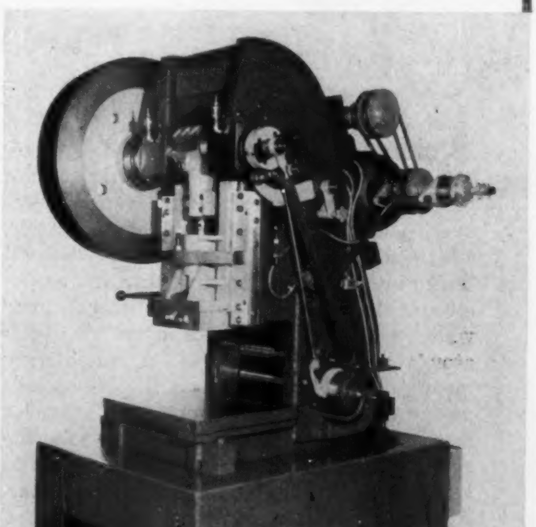
Automatic Stamping Press

SPEED is infinitely adjustable from 90 to 350 strokes per minute for high-speed automatic stamping operations with a new 50-ton press developed by Precision Welder and Flexopress Corp. Use of an automatically controlled variable-speed drive permits a positive pin type clutch to be engaged at low speed, with speed automatically building up to a preset figure. On declutching, the variable-speed drive is released and the press returns to low speed.

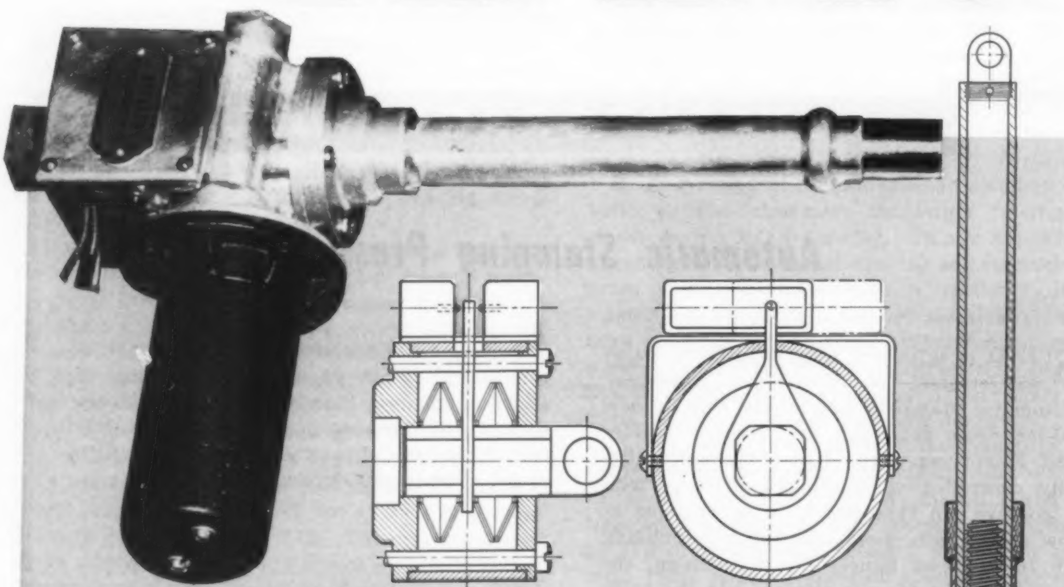
Speed control is obtained with a double set of Speed Selector Inc. variable-pitch sheaves, actuated by an air cylinder. Stroke of the cylinder is limited by stopnuts on the threaded end of a double-ended piston rod, which limits the high speed by controlling pitch diameter of one pair of sheaves. Air pressure is controlled by a valve, actuated by throwing the clutch lever on.

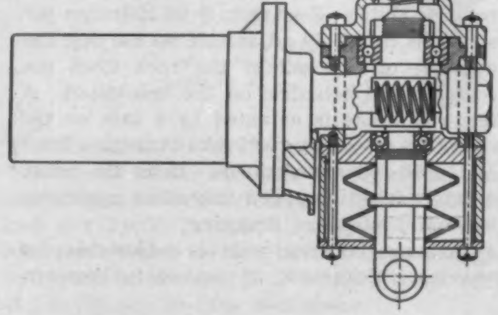
Coil stock is fed through the press by a pair of hardened feed rolls, intermittently operated by a rack and gear through an overrunning clutch. Feed from 0 to 15 inches per stroke is infinitely adjustable by varying the distance of the end of the rack from the center of an eccentric on the crankshaft. A feed-roll brake is actuated by a cam on the eccentric, operating the brake through a lever and push-rod arrangement. Thus, the brake is intermittently applied only when necessary, without continuous dragging.

Distance between rolls is adjustable for varying thicknesses of material; however, since four gears are used, the gear train from roll to roll is always in mesh regardless of feed-roll opening. Standard stroke is 3 inches with a $7\frac{1}{2}$ horsepower motor. A special 8-inch stroke model is available, however, with speed of 50 to 150 strokes per minute.



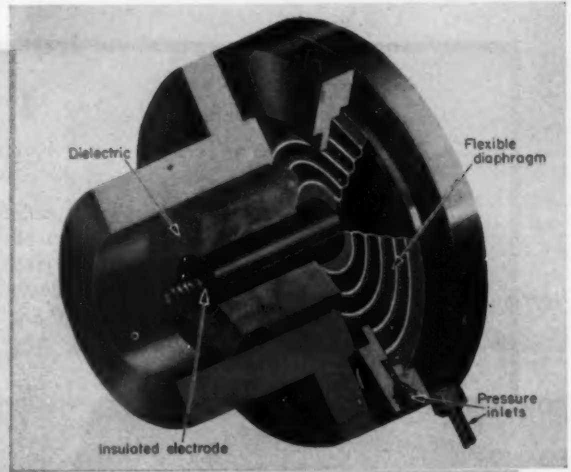
SCANNING the field for IDEAS



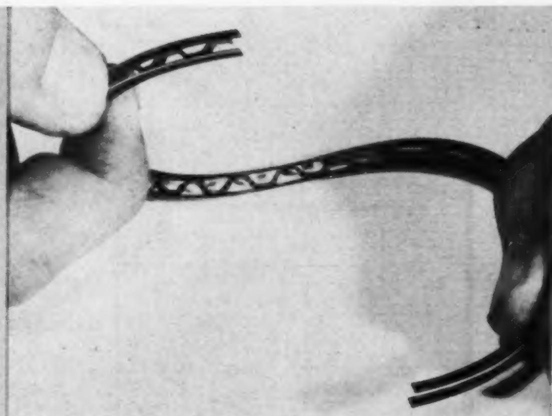


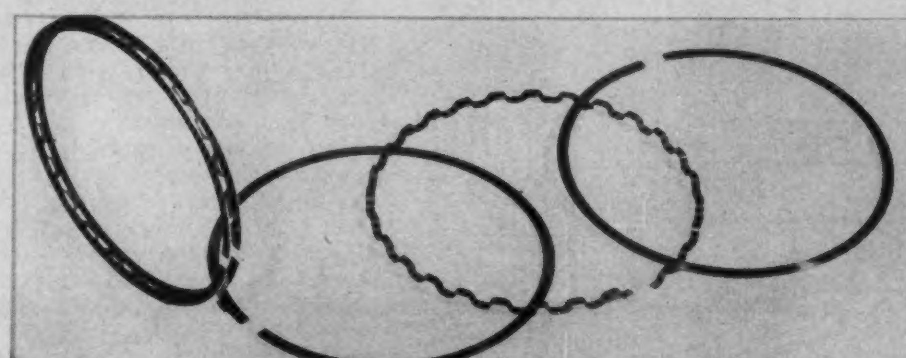
AUTOMATIC LOAD SENSING to prevent overtravel of a mechanical linear actuator is achieved with a limit switch system employing belleville springs. In a design developed by Excelco Developments Inc., overloads on the drive tube, in both directions of stroke, actuate a short stub shaft, compressing a series of belleville springs and operating a limit switch which is connected to the power source. Movements of the stub shaft are restricted to a lateral direction by an octagon design which prevents turning. In operation, a 0.060-inch travel of the actuator body is required to trip the switch under 125 per cent overload. A modification of the unit employing coil springs has also been developed; however, the flat load-deflection characteristics of the belleville spring types at certain loads have proved to be particularly suitable for most applications. A rugged, compact construction is provided by the design which assures dependable operation at temperatures varying from −65 to 165 F.

MINUTE DIFFERENTIAL PRESSURES as small as 0.000016-inch of water are precisely measured by a unique micromanometer developed by the National Bureau of Standards. Sensing element of the device is a corrugated diaphragm, 3 inches in diameter and made of 0.001-inch thick brass sheet, which is mounted opposite an insulated electrode to form a variable capacitor. Pressure or temperature changes in either of two pressure-tight cavities on each side of the diaphragm produce a flexing action, changing the electrode-to-diaphragm spacing and hence the capacitance value. Detection of capacitance variations, which are directly related to pressure and temperature differentials, is accomplished through a capacitance-type pickup and a resonant bridge carrier system. Providing remarkably high resolution and sensitivity characteristics, the device can be used for measuring static differential pressures, for sensing pressures varying at rates up to 20 cps with 2 per cent accuracy or, in modified form, for measuring absolute pressures below 1 micron Hg.



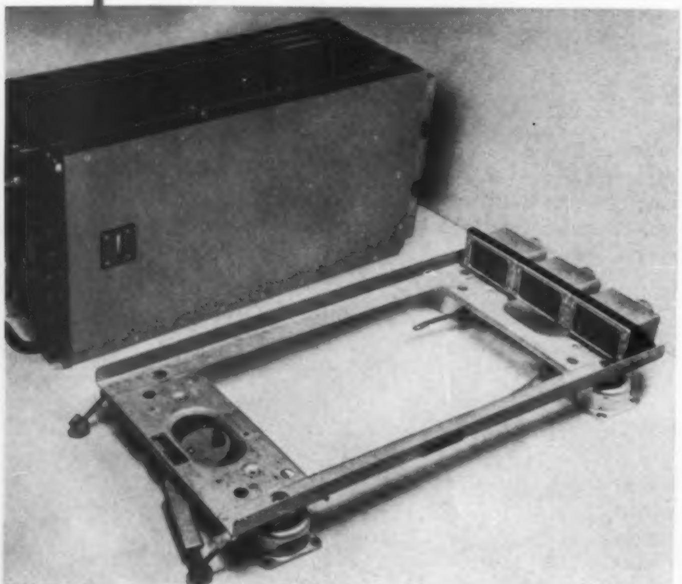
UNITIZED DESIGN of multi-piece piston rings offers a compact bonded assembly which can be installed as a single unit. Engine operation causes a special adhesive bond to dissolve, permitting the ring to separate into its independently operating parts. Developed by Muskegon Piston Ring Co., the three-piece oil control ring design employs a rubber-base flexible adhesive that is oil soluble to join two steel rails and a corrugated steel spacer in a one-piece unit assembly. Test results show that the adhesive completely dissolves within 300 to 500 miles of driving and is noninjurious to engine parts.

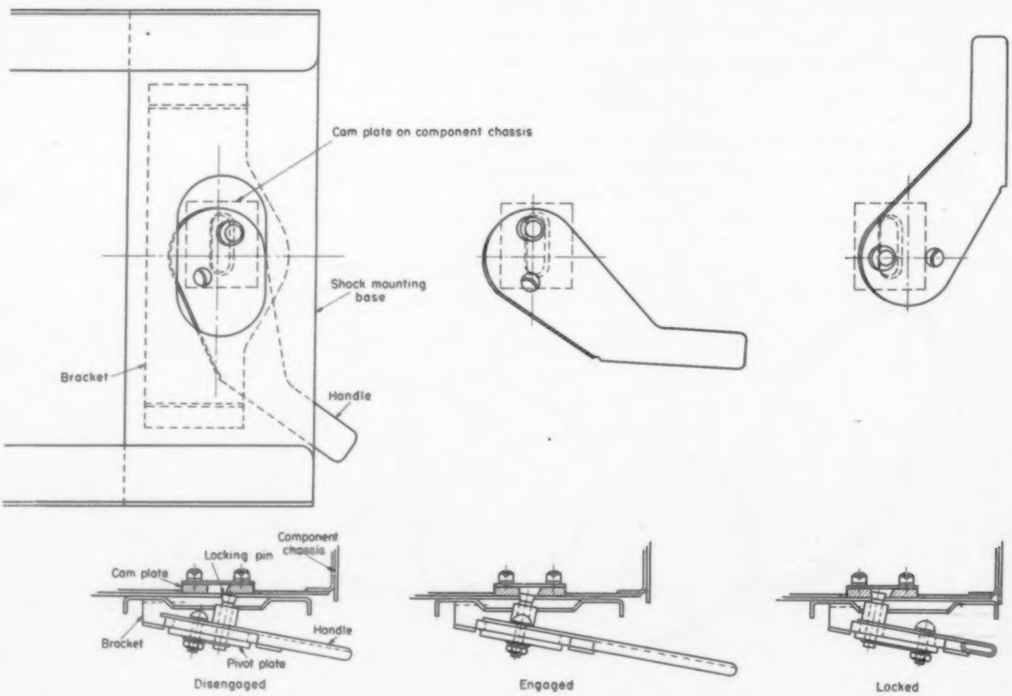




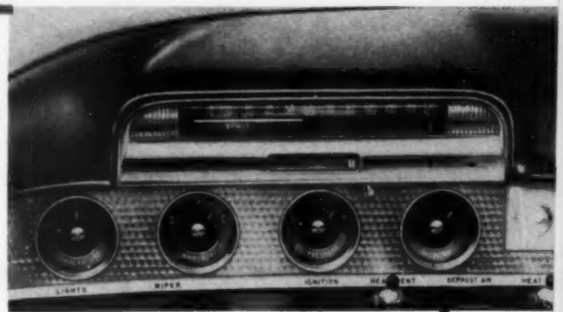
IDEAS

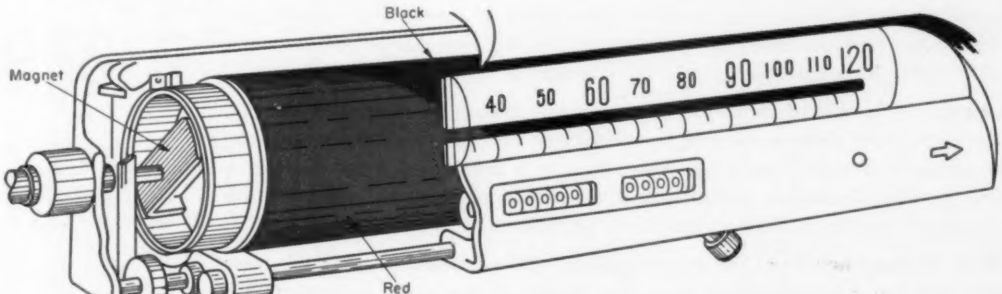
ACCURATE LOCKING for position, and retention of electronic assemblies under shock and vibration, are provided by a novel latch mechanism in a new rack mount design developed by Designers for Industry. High locking and release forces are produced by a lever-operated linkage, mounted at a 15-degree angle with the rack platform, which has a mechanical advantage of approximately 8/1 and is actuated through a handle extension. Lever movements are transmitted to mounted equipment by a pin projection which moves through an inclined arc to engage and disengage a cam plate on the bottom of the electronic assembly chassis. Positive locking action and correct alignment of electrical connections are assured by plug connectors, attached to the rear of the operating assembly, which fit into mating receptacles on the rack as the mounted unit is moved forward into the locked position by lever action. Disengagement is accomplished by reversing the handle movement, providing a positive release of the plug connectors and permitting the mounted assembly to be lifted off the rack for inspection and maintenance. Developed for use with airborne and automotive equipment, the design reduces space requirements to a minimum.





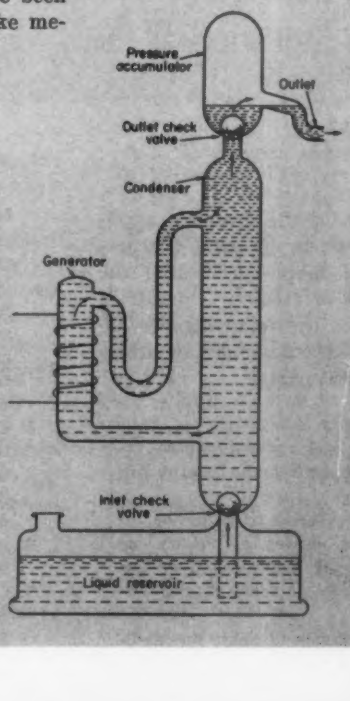
INSTANTANEOUS SPEED INDICATION in the new Buick "Redliner" speedometer is obtained by means of a novel rotating drum design. Controlled through a magnetic drive, the drum indicates speed through a solid horizontal red line which seems to flow across a graduated scale, eliminating the conventional needle. The illusion is created by a diagonal red and black color pattern on the drum. As the car accelerates, rotation of the drum turns up the red pattern which appears as an advancing line on the dial; deceleration reverses the drum rotation, causing the red line to recede.

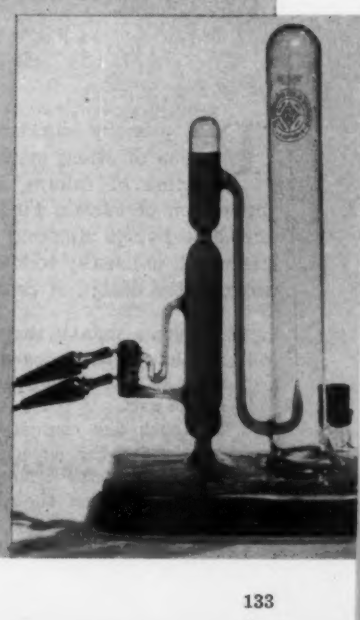




THERMAL FLOW CONTROL in a new heat-operated pump design minimizes moving parts and eliminates noise and wear. The Thermo-Pump developed by Jet-Heet Inc. is operated by either electric or flame heat and is based on a principle comparable to that of the coffee percolator. Pressures of 25 psi and flow rates up to 8 gph of fuel oil have been obtained with small model units. Unlike mechanical pumps, the thermal units can be constructed entirely of glass and hermetically sealed for use with dangerous or corrosive liquids and chemicals.

Pumping action is obtained through a sequence of vaporizing and condensing operations. Fluid is vaporized in a heated vapor generator and bubbles through a tube to collect in a condenser where pressure is built up. Pressurized vapor leaves the condenser through a check valve, is condensed and passes through an outlet jet to the external circuit. Fluid circulation is continuous; the flow of vapor creates a hydrostatic unbalance, forcing cool fluid into the generator and simultaneously operating a check valve to draw in new fluid from a reservoir. Although efficiency is lower than for comparable mechanical pumps, the design offers advantages where there is a source of waste heat or where electric and mechanical power are either unavailable or inconvenient.





Increasing demands for accurate knowledge of the strength of machine components has been paced by corresponding developments in new and improved techniques which aid design for strength.

Early progress in design was characterized by the exercise of the designer's native talent, his command of a few fundamental physical relationships, large ingredients of experience—and for good measure, his judicious selection of safety factors that more often covered the "unknowns" in design rather than "safety" only.

There is little need to point to any of the myriad product requirements and economic influences that have led to demands for maximum efficiency in the use of materials in design, absolute prevention of premature failure, and other requisites that are being met by use of the full complement of methods available today for accurately predicting strength in service.

Commonly identified in recent years by the broad label *stress analysis*, the whole body of theoretical and experimental concepts and their application in practical design have been developed to a high level. But the gains that are thereby offered in design are often offset by personal unfamiliarity with available methods, their virtues and limitations.

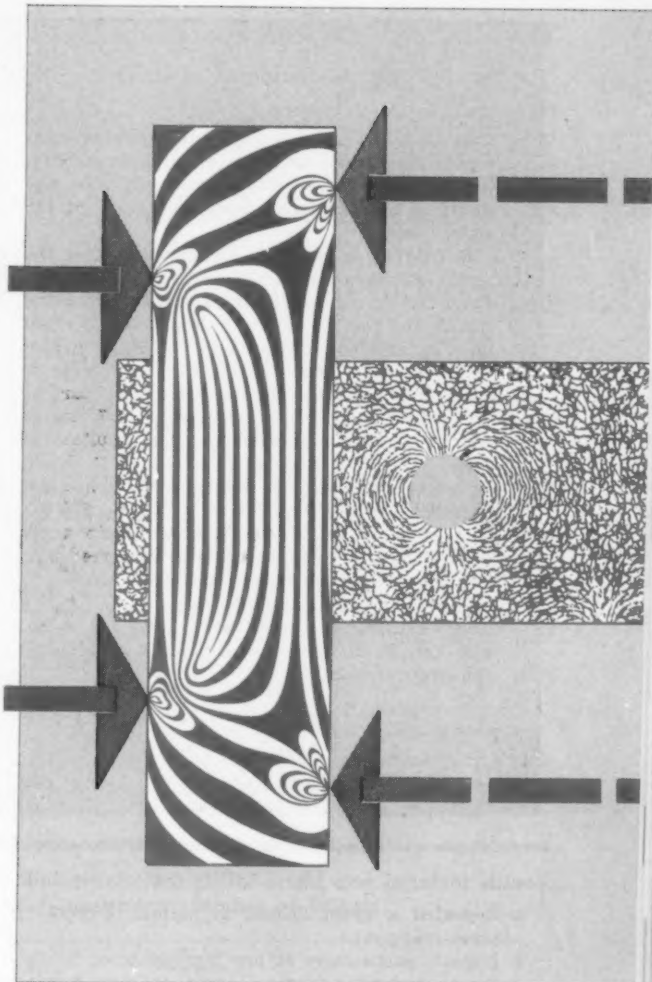
True, the way has been led by specialists within particular areas of stress analysis and the methods often seem far advanced for easy command and common use. Just as in all design, however, some problems are easily solved, and at the other end of the scale there remain or are being uncovered those that defy ready analysis. Gains to be gathered, however, seem porportional to the designer's interest and pursuit of the subject—and to his judgment in matching the design situation with the proper approach.

This series of articles is designed as a guide to stress analysis—not as a text. The literature consisting of current texts, periodicals and society proceedings, many of which are referenced in these articles, provides the basic tools. In this discussion an attempt will be made to formulate a comprehensive view of the problem, compare the various theoretical and experimental approaches available, and propose a general plan of attack for design. The articles will give no easy answers to specific problems; they will help the designer's search for the best approach to the answer.

THE primary objective of the practical application of stress analysis in design is the prevention of failure, or more specifically, the prevention of certain kinds of failure. Pursuit of this objective is apparent in the following procedure which is usually followed, consciously or otherwise, in the design of critical parts:

1. Develop a suitable shape.
2. Predict probable loads and environmental effects to be encountered during the useful life of the part.
3. Establish and evaluate *criteria of behavior,* in relation to the expected *modes of failure,* as measures of the effects of loading and/or environment upon the part (stress, strain, deflection, etc.)
4. Select materials on the basis of their mechanical, physical and chemical properties—and their economy. Selection involves a comparison between the anticipated effects of loading and environment, expressed in terms of pertinent *criteria of behavior,* (such as stress), and corresponding limiting characteristics of the material (such as yield or ultimate strength). Also to be considered are expected life of the part and consequences of failure.
5. Choose a *theory of failure* in conjunction with the foregoing comparison. Choice may depend upon the type of material (brittle or ductile), the nature of the stress, or accepted practice dictated by codes.
6. Select final dimensions, tolerances, surface finish, etc.
7. Construct prototype, test, appraise performance, and if necessary, redesign in the light of test results.

Stress Analysis in Design



By J. B. Hartman and R. E. Benner
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Part 1–Fundamentals and Theoretical Methods

In this outlined rational design procedure, the three italicized terms identify fundamental ideas that might best be first defined and discussed.

Mode of failure means simply the anticipated manner in which a part will fail. Since these articles will deal with the prevention of structural failure of parts, the possible variety of modes of failure is reduced to just three: (excessive) elastic deflection, yielding, and fracture. And although methods of stress analysis are helpful in preventing excessive elastic deflection, the only modes of failure to be considered here will be yielding and fracture.

Mode of failure is influenced by such factors as

1. Type of loading, i.e., static, repeating.
2. Duration of the load.
3. Nature of material (ductile or brittle).
4. Shape of part (effect of geometry on stress concentration).
5. Environmental effects, such as moisture.
6. Temperature (creep at high temperatures, brittle behavior below transition temperature).

The distinction between *ductile* and *brittle* behavior should be clearly understood. Actually the nature of the behavior is as much dependent upon the imposed conditions (such as type of loading, state of stress, temperature, strain rate) as upon the nature of the material. Thus a material such as steel which behaves in a ductile manner when subjected to gradually applied loads may experience brittle failures (i.e., act like a brittle material) when subjected to impact loads at low temperatures. The terminology of "brittle" or "ductile" is useful in denoting the ability of a material to either yield appreciably or fracture before the occurrence of appreciable yielding when circumstances are conducive to yielding.

Some of the conditions under which a normally

Fig. 1—Definition of stress

Stress is an internal force per unit area tending to resist size or shape changes in a body. Its intensity is dependent upon the size and shape of the body and upon the nature of the acting loads.

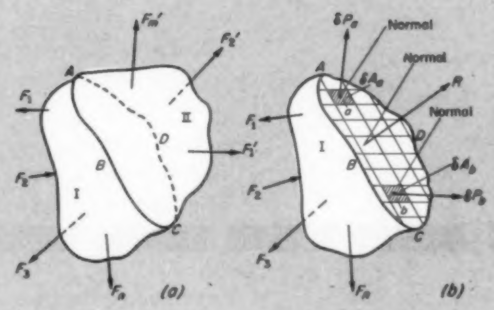
Body at *a* is in static equilibrium under the action of external forces $F_1, F_2 \ldots F_n$ and F_1', $F_2' \ldots F_m'$. If the body is parted at plane *ABCD* and part II removed, then part I is acted upon by external forces $F_1, F_2 \ldots F_n$. Provided these external forces do not form a system in equilibrium, it must be concluded that part II originally exerted, internally on part I, forces which maintained equilibrium. The resultant of these forces is denoted by *R* in view *b*. It should be observed that the resultant need not act normal to plane *ABCD*.

If cross-section *ABCD* is divided into small areas δA_a, δA_b, etc., there will be forces δP_a, δP_b, etc., acting on each area such that

$$\Sigma P \ \delta A = R$$

Then, consider one such area, δA_a, for example. Let it diminish about point *a* and define the stress at point *a* as

$$s_a = \lim_{\delta A_a \to 0} \frac{\delta P_a}{\delta A_a}$$



In the most general case, stresses may vary from point to point within a body, or for example, $s_a \neq s_b$ in view *b*.

Stress is a tensor quantity; it can be represented by a vector of magnitude and direction dependent upon the orientation of the plane through the point in question. If planes other than *ABCD* were passed through point *a*, for example, s_a would have a different magnitude and direction. To completely specify stress, one must give the magnitude and direction of the stress vector and the orientation of the plane upon which the stress is acting.

These fundamental factors are discussed in more detail in the Appendix of this article.

ductile material acts like a brittle material include

1. Repeated or cyclic loading at normal temperatures (fatigue).
2. Impact, particularly at low temperatures.
3. Creep (extended static loading at high temperatures).
4. Triaxial state of stress, as at the bottom of a deep groove.
5. Severe quenching without tempering.
6. Strain hardening accompanying yielding.

Criterion of behavior is that quality or attribute of the loaded part that is the most pertinent measure applicable to a selected mode of failure. Just as there may be many modes of failure in a variety of situations, so might there be several criteria of behavior for each mode of failure. Some of these possible combinations are suggested[1] in Table 1. If discussion is limited to yielding and fracture, as modes of failure, *stress* then is the criterion of failure.

Since *stress* is now established as the central idea of much of the discussion to follow, it should perhaps be well defined also. Its "technical" definition for the purposes of these articles is given in *Fig.* 1.[2]

Ideally the purposes of a stress analysis are to predict the manner in which a machine part is likely to fail and to devise a means of determining the ability of the material to withstand the expected type of failure. These purposes have been joined in procedures which are based upon *theories of failure*.[3] Traditionally each of the various proposed theories of failure has named a significant quantity as being the cause of failure, said to occur when the value of that quantity reaches a critical value, the magnitude of which is determined in a simple tensile test.

It would indeed be fortunate if one theory of failure could be applied universally. Such a theory is not available because of the possible differences in the mode of failure, in addition to the simplifying restrictions on the method of testing the material as in a simple tensile test.

The best known theories of failure which have been applied to cases where failure is considered to occur, due to yielding or fracture under static loading, are identified and described in Table 2.

It is important to note that once it has been established that a given theory of failure yields valid results, critical conditions in a structural member may be ascertained without resorting to experiments with infinite combinations of stresses. The theories discussed in Table 2 define failure in terms of yielding or fracture under static loading. Unfortunately the designer has other conditions to consider. For example, there are many instances in which inelastic strains occur locally in machine parts without failure of the whole piece. Members made of ductile materials subjected to simple static loads at ordinary temperatures are under certain circumstances capable of local plastic deformation which results in a redistribution of stresses. A more favorable distribution of stress may result, permitting an increase in load without overall struc-

[1]References are tabulated at end of article.

tural failure. The regions where this inelastic action may occur without failure are generally near holes, notches or grooves where there is a large stress gradient.

A real restriction in the applicability of the theories of failure for the machine designer lies in the fact that rarely do machine parts fail because of unexpectedly heavy static loads. Serious breakage of machine parts are usually caused by repeated or fatigue loads. Failures may occur without warning beginning with localized action at points of stress concentration unaccompanied by plastic deformation. The limiting values of the alternating stresses as determined by various theories of failure must be expressed in terms of the endurance limit determined by repeated-load (fatigue) tests rather than tensile tests used for static loading. In the majority of cases encountered in machine design, alternating stresses are superposed on stress components that are static. Present procedures involving ductile materials subjected to combined steady and alternating loads make use of graphical representations such as the Goodman diagram to determine working stresses.

So far this discussion has dealt with certain fundamental ideas and their relationship in a rational design procedure. As evident from this approach—and as intuitively understood commonly in design—*stress* is the factor upon which attention must be focused.

STRESS ANALYSIS IN DESIGN

In the remainder of these articles procedures of stress analysis will be examined and compared with particular regard for their usefulness in design. These methods are, for convenience, summarized in advance in TABLE 3.

I THEORETICAL STRESS ANALYSIS

Machine design begins with pencil and paper. A careful and realistic theoretical analysis can produce worthwhile savings in money and time. Possible failures may be predicted and avoided before the design is constructed.

Two basic steps are involved in determining mechanical strength by the rational methods of theoretical stress analysis:

1. Evaluate those stresses within a member which will be significant criteria of behavior.
2. Establish maximum allowable stress conditions.

The second step provides a criterion with which the stresses of the first step may be compared so

Table 1—Significant Criteria of Behavior for Various Modes of Failure

Modes of Failure	Usual Criteria of Behavior	Examples
Elastic Deflection		
Stable equilibrium	Linear or angular displacement	Torsional deflection of cam shaft may provide undesirable alteration in timing of controlled event. Even though stresses may not exceed the elastic limit, the shaft fails to perform properly. In general, excessive deflection under given load can be remedied by changing shape or dimensions of member or by selecting stiffer (not necessarily stronger) material.
Unstable equilibrium (instability)	Comparison of applied load with critical value of load	Buckling of long slender bars subjected to axial compression or buckling of thin plates subjected to edge compression can occur at stresses lower than the elastic limit. Critical values of load may be accompanied by excessive elastic deflections. Loads exceeding critical values may lead to total collapse of member.
Vibration	Amplitude, frequency, transmissibility	Gear trains are sometimes unsatisfactory due to noise of accompanying vibrations which result from poor tooth action. Unbalanced machinery may have to be "isolated" from supporting foundations to reduce the "transmissibility." In these cases, the mere presence of vibration causes unsatisfactory performance.
Plastic Deformation		
Yielding	Stress	Materials exhibiting a yield point are usually considered to have failed whenever the plastic region is extensive enough to allow large (permanent) deformations of the member. Various theories of failure have been proposed which attempt to predict those combinations of stress which cause yielding.
Creep	Stress	Some materials, notably metal and plastics, deform continuously with time when subjected to constant stress and "elevated" temperatures. Designing for creep may involve choosing a working stress which will produce a deformation at the operating temperature not in excess of some limiting value throughout the expected life of the member.
Fracture		
Brittle (static load)	Stress	Brittle fracture denotes a sudden failure in which the member breaks into pieces with little or no plastic flow preceding the separation. High-carbon steel, glass and gray cast-iron behave as brittle materials at "normal" temperatures.
Fatigue	Stress	A fatigue failure results from the application of repeated loads to a member. Fracture begins with a small crack which gradually spreads until the member suddenly breaks. Working stresses must be chosen which are compatible with the number of hours service desired.

that failure will be avoided and economy of material assured.

The ability of theoretical methods to accomplish step 1 is limited by various considerations. The following discussion of some of the methods and limitations of theoretical stress analysis will indicate certain factors involved in judging its usefulness and reliability in evaluating significant stresses.

Development of Theoretical Formulas: Both *strength of materials* and the classical mathematical *theory of elasticity* are theoretical methods of use to the designer in his attempt to predict stresses resulting from external loads. Each method produces formulas with which the size and, to an extent, the shape of members may be so determined that strength requirements are satisfied. Although the theory of elasticity is more rigorous and more generally applicable than is strength of materials, the general sequence of development is similar for both. This sequence may be summarized briefly in four steps:

1. Development of equations of equilibrium, or Development of equations of motion.
2. Geometrical analysis (strain).
3. Stress-strain relationships.
4. Combination of first three steps for solution of a particular problem. (Development of a formula which will yield a numerical solution.)

Strength of materials and elasticity differ with respect to the way each of these steps is carried out. Each step will be discussed briefly and the two theoretical methods will be compared. Certain aspects which limit their range of application will be indicated. Pertinent aspects of each method are shown in *Fig.* 2. Frequent reference to this figure may be helpful in the following discussion.

Equations of Equilibrium: External forces may be either body or surface forces. Body forces are those which are proportional to the mass contained within the volume of a body such as gravitational or magnetic attraction. Surface forces are those which act upon the surface of a body such as concentrated or distributed loads acting upon a beam. If a body remains at rest under the action of external forces, it is in static equilibrium and the following conditions must prevail: (1) The external forces must be in equilibrium, and (2) any portion of the body must be held in equilibrium by the internal and external forces acting upon that portion. The second condition of equilibrium provides the principle upon which the equations of equilibrium are based.

In strength of materials, the body is cut by an imaginary plane passed through the point of interest in the desired direction and the resulting "freebody" portion is examined. A mathematical expression is developed which relates the stresses acting upon this cross-section to the external forces. In these equations of equilibrium, the stresses appear within integrals. Because these integrals cannot be evaluated without additional information, the stresses cannot be determined from the equations of equilibrium alone.

An elemental cube from the interior of the body is analyzed in the theory of elasticity. Stresses and body forces acting upon this cube are related by a series of partial differential equations which are developed upon the premise that static equilibrium of the cube maintains. Because surface forces do not act upon interior portions of a body, they do not appear in these equations; however, they do influence the stress field and must be considered. Their restrictions on the stress field would be made evident in some of the so-called *boundary conditions.* Boundary conditions, in general, are mathematical statements describing the physical conditions to which the body involved is subjected for a particular problem. A boundary condition which is often difficult to meet is that the components of stress at the boundary must equal the components of the surface forces per unit area of boundary.

The equations of equilibrium developed in elasticity are of a general nature and are utilized in the solution of many different types of problems. Specific types are identified by specification of bound-

Table 2—Theories of Failure

Theory	Description
Maximum Principal Stress	Holds that inelastic action begins at a point in the material when the maximum principal stress at that point reaches the elastic limit, as found in a simple tensile (or compressive) test. Theory has proven reasonably satisfactory for brittle materials only.
Maximum Strain	States that inelastic action begins at a point when the maximum strain at the point reaches the value attained at the elastic limit in a simple tensile (or compressive) test. This theory, too, is effective primarily where failure is due to brittle fracture.
Maximum Shearing Stress	Compares maximum shear stress on some plane through the point with the maximum shear stress existing at the beginning of inelastic action in a simple tensile test. This theory has been applied successfully for ductile materials in widely used machine parts and forms the basis of the ASME code for transmission shafting.
Total Strain Energy	Maintains that inelastic action at a point in a body is initiated when the energy per unit volume to be withstood at the point in question equals the energy absorption at the elastic limit in a simple tensile test. Tests on materials at high hydrostatic pressures cannot be explained by this theory.
Distortion Energy	Resulted from shortcomings of the total strain energy theory. This theory holds that inelastic action begins at a point subjected to any combination of stresses provided the strain energy of distortion per unit volume reaches the level attained at the elastic limit in a simple uniaxial test.
Octahedral Shear Stress	Yields identical results with the distortion energy theory although the results are obtained in terms of stress rather than energy. The distortion energy theory or its equivalent, the octahedral shear stress theory, seems to produce the best results for statically loaded ductile metals, although it is slightly less conservative than the maximum theory of failure.

ary conditions and body forces. The equations of equilibrium associated with strength of materials relate stresses to the external forces involved and therefore are valid only for the specific type of problem for which they were developed such as pure tension or flexure.

Equations of Motion: Formulas developed using the assumption that static equilibrium conditions prevail are valid only for statically applied loads. A load may be presumed to be static if its magnitude, point of application, and direction do not change with time. A beam acted upon by its own weight or by the weight of a wall it supports are examples of static loads.

A dynamic load is an externally applied force whose magnitude, point of application, or direction varies with time. All three force characteristics may vary individually or in combination. Accelerations of a body will result whenever dynamic loads are applied. These accelerations have the effect of introducing additional forces which in turn affect the resultant stress system, possibly producing higher stresses than if loads of the same magnitude and distribution were applied statically. It is apparent that static formulas may yield incorrect results when used to analyze dynamic systems.

D'Alembert's principle offers a convenient method whereby the effects of acceleration may be considered in the development of equations relating stress to external load. Thus, by considering the inertia force components as additional body forces and introducing them into the equations of equilibrium of the theory of elasticity, the general equations of motion applicable to dynamic systems are obtained. It should be recognized that this system may break down if the acceleration is so high that shock (impact) conditions are likely to prevail in the member under analysis.

General equations of motion relating stress to external load are not usually set up explicitly in strength of materials. However, from vibration theory in which equations of motion are usually developed in terms of displacements, it is sometimes possible to calculate deflections of the dynamically loaded system. This permits a *dynamic load factor* to be evaluated.[4] The dynamic load factor may be defined as the ratio of maximum deflection under dynamic conditions to the deflection the system would experience if the peak dynamic load were applied statically. Multiplying the peak dynamic load by the dynamic load factor will yield the so-called *equivalent static load*. This procedure makes possible the calculation of dynamic stresses by utilizing the static formulas of strength of materials together with the equivalent static loads. This method is further amplified[4] by the example shown in *Fig.* 3.

Although the equations of equilibrium or of motion relate stress to external load, they provide insufficient information for the calculation of stress. As a result, it becomes necessary to investigate the geometry of elastic deformation so that additional information may be obtained.

Geometrical Considerations: Whenever an elastic body is subjected to load, it deforms in a manner dependent upon the shape of the body and the distribution of load. In strength of materials, the most common procedure is to make assumptions concerning the manner in which a particular body will deform when subjected to a given load distribution. These assumptions are usually supported by experimental evidence. Analysis of the geometry of the deformed member provides an expression in which strain distribution is related to pertinent dimensional parameters of the body. The equations thus obtained are valid only for the shape and load distribution of a specific type problem, such as torsion of a circular shaft.

A geometrical analysis is also made in the theory of elasticity. However, the approach and resulting equations are more general. From an investigation of the geometry of a deformed differential element, equations relating unit strain to displacements of points within the body are obtained. Also in recognition of the fact that strains within a continuous medium are not independent of each other, a series of *equations of compatibility* are developed which interrelate the various normal and shear strains. These equations are general in nature and apply to many types of problems. Unlike strength of materials, prediction of strain distribution is not necessary in the theory of elasticity.

Stress-Strain Relationships: It has been mentioned that the compatibility equations show a necessary strain relationship for a continuous medium. For an elastic material, stress is uniquely dependent upon elastic strain. Therefore, it becomes

Table 3—Stress Analysis Procedures

A. Theoretical

1. **Strength of Materials.** Also called mechanics of materials, this method utilizes equilibrium of forces and simplified assumptions concerning strain.
2. **Theory of Elasticity.** A more detailed mathematical method in which conditions of equilibrium are satisfied, the complete distribution of strains is taken into account, and a more general statement of Hooke's law is applied.
3. **Theory of Plasticity.** Concerned with behavior of ductile materials beyond elastic range. Theory undergoing rapid development.

B. Experimental

1. **Strain Gages.** Local measurements over specific gage lengths on surface.
   a. **Mechanical**
   b. **Optical**
   c. **Electrical**
2. **Brittle Lacquers.** Provides an overall picture of strain distribution on surface.
3. **Photoelasticity.** Provides picture of stress distribution through interior of model.
4. **Analogies.** Methods of analysis providing interchangeable experimental approaches because of similarity of mathematical equations involved.
5. **X-ray Analysis.** Measurement of spacing between planes of atoms.

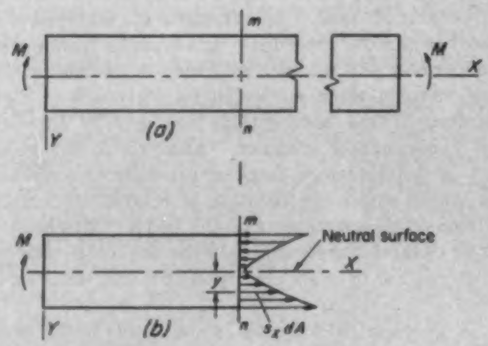
C. Empirical. Utilization of experience, sometimes without rationalization.

D. Combinations of A, B and C

STRENGTH OF MATERIALS

THEORY OF ELASTICITY

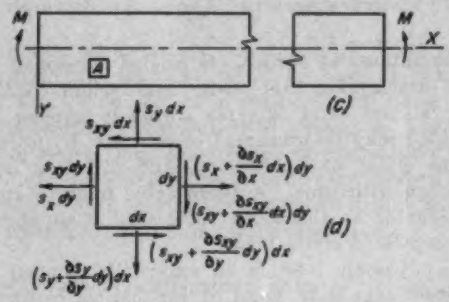
1—EQUATIONS OF EQUILIBRIUM



To find the stresses on plane mn resulting from bending moment M, analyze free body b, where stress at the neutral surface $s_x = 0$, dA = elemental area, and y = distance from neutral surface to force $s_x\,dA$. Then, by summing moments

$$M = \int_A s_x\, y\, dA \qquad (2.01)$$

Note that various distributions of stress s_x could satisfy Equation 2.01. More information is required for determination of the actual distribution.



The free-body diagram of cube A extracted from an interior portion of beam c, subject to bending moment M, is shown at d. Forces acting on each face are indicated. Because two dimensions only are of concern, let $dz = 1$ for convenience. Body forces will be neglected. Summing forces in the X and Y-directions yields the equations of equilibrium:

$$\left.\begin{aligned} \frac{\partial s_x}{\partial x} + \frac{\partial s_{xy}}{\partial y} = 0 \\ \frac{\partial s_y}{\partial y} + \frac{\partial s_{xy}}{\partial x} = 0 \end{aligned}\right\} \qquad (2.10)$$

These equations do not provide sufficient information for the solution of stresses.

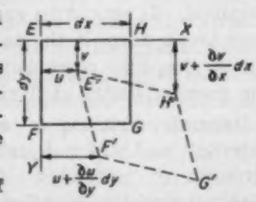
2—GEOMETRICAL ANALYSIS

The assumption is made that plane sections normal to the longitudinal axis of the beam, such as mn, remain plane and normal to the axis after bending. From this assumption it follows that

$$\epsilon_x = \frac{y}{r} \qquad (2.02)$$

where ϵ_x = strain of a fiber at distance y from the neutral surface and r = radius of curvature of neutral surface of beam after bending. Equation 2.02 states that strain ϵ is proportional to distance y.

Fig. 2—Solutions for stress in a beam subjected to pure bending by the methods of *strength of materials* and *theory of elasticity*



Consider element A in the strained condition $E'F'G'H'$. Symbols u and v represent components or displacement of point E in the X and Y-directions.

Displacement of point H in the X-direction is $u + (\partial u/\partial x)\,dx$. Hence, strain in the X-direction is

$$\epsilon_x = \frac{\left(u + \dfrac{\partial u}{\partial x}\,dx\right) - u}{dx} = \frac{\partial u}{\partial x} \qquad (2.11)$$

Similarly, strain in the Y-direction is

$$\epsilon_y = \frac{\partial v}{\partial y} \qquad (2.12)$$

Shear strain γ_{xy} = angle HEF − angle $H'E'F'$. Therefore,

$$\gamma_{xy} = \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \qquad (2.13)$$

The relationship among the strains is exhibited by the compatibility equation:

$$\frac{\partial^2 \epsilon_x}{\partial y^2} + \frac{\partial^2 \epsilon_y}{\partial x^2} - \frac{\partial^2 \gamma_{xy}}{\partial x\,\partial y} = 0 \qquad (2.14)$$

STRENGTH OF MATERIALS

THEORY OF ELASTICITY

3—STRESS-STRAIN RELATIONSHIPS

Strength of Materials

Stress distribution is determined from strain distribution by use of stress-strain relationships. Or, by simplified statement of Hooke's law,

$$s = \epsilon E \qquad (2.03)$$

where E = modulus of elasticity. For this problem,

$$\epsilon_x = \frac{s_x}{E} \qquad (2.04)$$

Combining Equations 2.02 and 2.04 gives

$$s_x = \frac{Ey}{r} \qquad (2.05)$$

which states that s_x is proportional to y.

Theory of Elasticity

A more generalized form of Hooke's law is

$$\left.\begin{aligned} \epsilon_x &= \frac{(s_x - \mu s_y)}{E} \\ \epsilon_y &= \frac{(s_y - \mu s_x)}{E} \\ \gamma_{xy} &= \frac{2(1+\mu)}{E} s_{xy} \end{aligned}\right\} \qquad (2.15)$$

where E = modulus of elasticity and μ = Poisson's ratio.

Substituting Equations 2.15 in Equation 2.14 gives the compatibility equation in terms of stress:

$$\frac{\partial^2 s_x}{\partial y^2} - \mu \frac{\partial^2 s_x}{\partial x^2} + \frac{\partial^2 s_y}{\partial x^2} - \mu \frac{\partial^2 s_y}{\partial y^2} - 2(1+\mu)\frac{\partial^2 s_{xy}}{\partial x \partial y} = 0 \qquad (2.16)$$

4—SOLUTION

Strength of Materials

Replace s_x in Equation 2.01 by its equivalent from Equation 2.04:

$$M = \int_A \frac{E}{r} y^2\, dA = \frac{E}{r} \int_A y^2\, dA \qquad (2.06)$$

By definition

$$\textstyle\int_A y^2\, dA = I \qquad (2.07)$$

where I = moment of inertia of A with respect to the neutral surface. Therefore

$$M = \frac{EI}{r} \qquad (2.08)$$

Combining Equations 2.05 and 2.08, and rearranging,

$$s_x = \frac{My}{I} \qquad (2.09)$$

which is the desired result. With little additional work, it can be shown that the neutral surface passes through the centroid of section mn.

Theory of Elasticity

Stress function $\phi(x, y)$ is defined as

$$\left.\begin{aligned} s_x &= \frac{\partial^2 \phi}{\partial y^2} \\ s_y &= \frac{\partial^2 \phi}{\partial x^2} \\ s_{xy} &= -\frac{\partial^2 \phi}{\partial x \partial y} \end{aligned}\right\} \qquad (2.17)$$

With these definitions Equations 2.10 are satisfied. Substituting Equations 2.17 into Equation 2.16 shows the compatibility relationship which $\phi(x, y)$ must satisfy:

$$\frac{\partial^4 \phi}{\partial x^4} + 2\frac{\partial^4 \phi}{\partial x^2 \partial y^2} + \frac{\partial^4 \phi}{\partial y^4} = 0 \qquad (2.18)$$

The stress function for this problem is known to be $\phi = ay^3/6$ (where $a = M/I$), which yields the solution:

$$\left.\begin{aligned} s_x &= ay = \frac{My}{I} \\ s_y &= s_{xy} = 0 \end{aligned}\right\} \qquad (2.19)$$

apparent that the compatibility equations also restrict stress. A relationship between stress and strain must be found (for each material of engineering interest) so that the equations of compatibility may be expressed in terms of stress, thereby exhibiting the stress relationships required to insure proper strain distribution. At the same time, the compatibility equations when expressed in terms of stress, provide the information needed, in addition to the equations of equilibrium or of motion, for solution of stress.

In strength of materials, the assumed strain distribution must be expressed in terms of stress so that the comparable stress distribution can be determined; therefore, stress-strain relationships are required here also.

Methods of Solving Specific Problems: Although both strength of materials and elasticity utilize the same general principles in their development, specific problems are solved somewhat differently by each method.

In strength of materials, the equations of equilibrium and strain are written. The latter may

then be expressed in terms of stress by use of the stress-strain relations. Proper combination of the resulting equations will yield a solution. The necessity of having to predict strain distribution in strength of materials provides an inherent difficulty and limits this method to the simpler problems.

If in elasticity, the differential equations of equilibrium or of motion and the compatibility equations were solved (subject to the pertinent boundary conditions) the result would be the desired equation for stress. Unfortunately, mathematics has not advanced to the point where solutions can be made in the more complex but practical cases. Therefore, more devious and less elegant methods must be used to obtain results.

The semi-inverse method credited to Saint-Venant involves making certain assumptions regarding the components of stress, strain or displacement. If it can be shown that the equations of equilibrium or of motion, the boundary conditions, and the compatibility equations are satisfied, then the solution based on these assumptions is the correct one. The difficulty here lies in making the correct assumptions commensurate with the problem at hand.

Another method of obtaining solutions is by use of the *stress function*. This function is defined in terms of stress components and is so defined that the equations of equilibrium are automatically satisfied. Further, it is possible to find many functions which satisfy the compatibility equations. Functions which satisfy both sets of equations are solutions of some problem. The task is to find the problem to which the solution applies. Many problems may be solved in this manner some of which are trivial while others are of no practical importance. Those of interest are available in standard references[5, 6, 7] on the subject. It is apparent that this method offers little hope whenever a fresh problem is to be solved because one must select the pertinent stress function which is almost tantamount to saying the answer must be known beforehand.

It is worthwhile to note that regardless of the

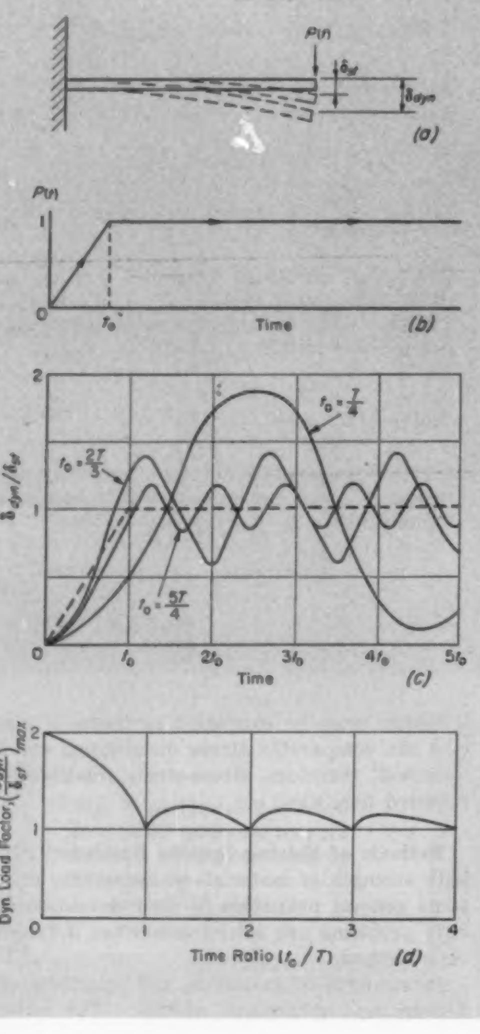


Fig. 3—Demonstration of dynamic loading of a cantilever beam

If the cantilever beam, *a*, responds as a single degree of freedom system, its strength under dynamic loading can be analyzed through the use of an equivalent static load. Let

$P(t)$ = Dynamic load expressed as a function of time
P_0 = Peak value of $P(t)$
T = Natural period of vibrating system
P_{eq} = Equivalent static load
δ_{st} = Deflection if P_0 is applied statically
δ_{dyn} = Deflection under dynamic conditions

One possible variation of load with time is shown at *b*. The initial load is zero and is gradually increased during the time t_0 after which it is maintained unchanged indefinitely.

When the load $P(t)$ is applied, the system will vibrate (in general) and the value of dynamic deflection δ_{dyn} will change with time. Graph at *c* shows the response δ_{dyn}/δ_{st} as a function of time for the loading shown at *b*. The response depends upon the ratio of time of application of load, t_0, to the natural period, T, of the vibrating system.

At *d* is shown the dynamic load factor for single degree of freedom systems subjected to the load pulse shown at *b*. The dynamic load factor is equal to the maximum value of the response for a given ratio, t_0/T; that is, $(\delta_{dyn}/\delta_{st})_{max}$. The equivalent static load is

$$P_{eq} = P_0 \left(\frac{\delta_{dyn}}{\delta_{st}} \right)_{max}$$

Curves differing from that shown at *d* result if the loading curve shown at *b* is changed.

method used, once a solution (which necessarily satisfies the equations of elasticity together with the boundary conditions) is obtained, that solution is not only a correct one but it is also unique. It is the only possible solution for the conditions of the particular problem, at least within the general limitations of the theory of elasticity.

The mathematical theory of elasticity is a more powerful method than is strength of materials and can be used to solve problems of greater complexity. Mathematical difficulties have limited its range of application; nevertheless, many practical problems have been solved and the method deserves attention.

APPENDIX—STRESS AND STRAIN FUNDAMENTALS

States of Stress: When all stresses acting on a small element of a body are parallel to one plane, the element is said to be in a state of two-dimensional or plane stress. This situation is sometimes designated as one of biaxial stress since the components of stress parallel to one of the three co-ordinate axes is zero. Numerous cases are encountered in design where the maximum stresses occur on the surface of a structural member. Usually the portion of the surface under consideration is unloaded, producing biaxial or plane stress. Hence it has important practical significance to the designer.

If the components of stress parallel to any two of the co-ordinate axes are zero, an axial (or uniaxial) stress condition exists.

When the point in question is subjected to stresses acting along the three co-ordinate axes, a state of tri-axial stress occurs, *Fig.* 4.

Biaxial and triaxial stress are sometimes referred to as states of combined stress. They have a considerable effect on the ductility and strength of the material.

Plane Stress at a Point: Being a vector quantity, stress must be defined in terms of magnitude and direction. Its value at a given point on a given plane is generally expressed in terms of a component normal to the plane, designated as normal stress, and a component parallel to the plane, designated as shear stress. Positive normal stresses denote tension—negative normal stresses denote compression. Shearing stresses acting on opposite faces of an element producing a clockwise couple are generally given a positive sign, with counterclockwise couples negative.

In the following analysis, attention to conventions of sign and nomenclature is necessary. Normal stresses parallel to a co-ordinate axis are written with a subscript corresponding to the axis. Thus s_x and s_y indicate normal stresses in the direction of the X and Y-axis, respectively. Shear stresses in the plane of the co-ordinate axes may be written with two subscripts. The first indicates the axis normal to the plane in which the shear stress acts. The second denotes the direction of the stress vector. Thus s_{xy} indicates shear in a plane normal to the X-axis acting parallel to the Y-axis. From considerations of equilibrium it is easily shown that s_{xy} is numerically equal to s_{yx}.

Fig. 5 represents a view of a prismatic element cut from a stressed body, with the stresses acting as shown. For plane stress, the dimension in the Z-direction is of no importance and will be considered as unity. The figure would then appear as a rectangular parallelepiped of unit thickness with finite dimensions in the X and Y-directions and with stresses s_x, s_y, and s_{xy} uniformly distributed over the faces. If the finite dimensions in the XY-plane were replaced with others of infinitesimal magnitudes dx and dy, the same method of analysis may be used for the case where the stresses are not uniformly distributed in the body. The sides of the element then become so small that s_x, s_y, and s_{xy} are essentially constant over the lengths of the element. By this extension of principles the equations which will be cited may be utilized to analyze stress at a point.

General Equations for Biaxial Stress: In order to investigate the general case of the resultant, or combined stresses acting on any inclined plane, the equations of equilibrium are applied to an elementary prism removed from the part in question and treated as a free body. Let the element be subjected to normal and shear stresses acting in directions as shown in *Fig.* 5. The normal to a diagonal plane makes an angle of ϕ with the X-axis. The resulting triangular prism may be treated as a free body in static equilibrium. Defining stress in terms of force per unit area, the total force on a face will be the product of its area and the stress acting. Area A of the inclined plane cancels out and the stress acting normal to the plane is

$$s_n = s_x \cos^2 \phi + s_y \sin^2 \phi - 2\, s_{xy} \sin \phi \cos \phi \quad \ldots\ldots (1)$$

If s_s represents the shear stress along the inclined plane, from equilibrium conditions

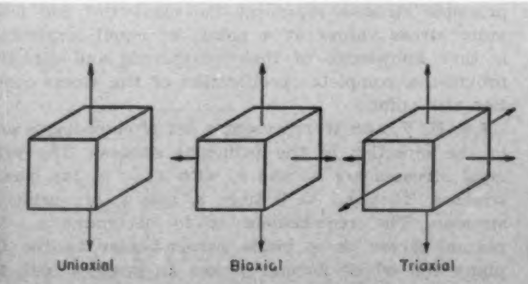


Fig. 4—States of stress

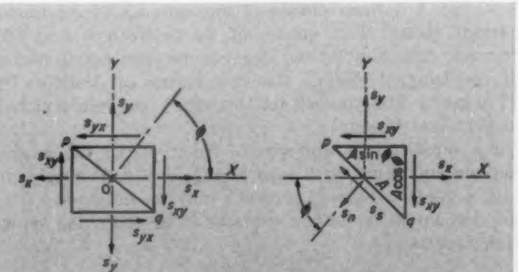


Fig. 5—Element subjected to normal and shear stresses

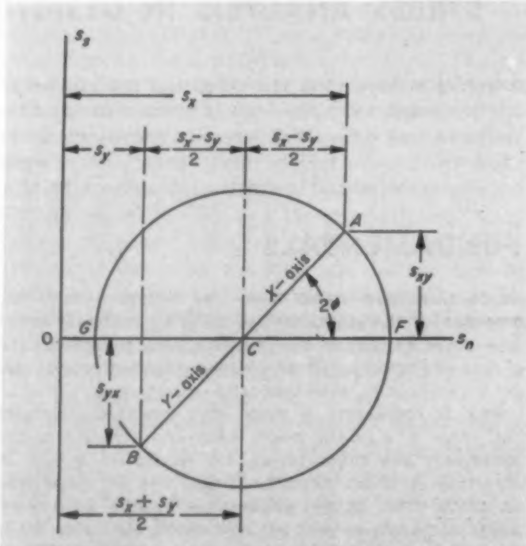


Fig. 6—Mohr's circle for graphically solving two-dimensional combined stress problems

$$s_s = (s_x - s_y) \sin \phi \cos \phi + s_{xy} (\cos^2 \phi - \sin^2 \phi) \quad \ldots (2)$$

It is often convenient to express these equations in terms of the double angle 2ϕ. By trigonometric substitution, the equations become

$$s_n = \frac{s_x + s_y}{2} + \frac{s_x - s_y}{2} \cos 2\phi - s_{xy} \sin 2\phi \quad \ldots (3)$$

$$s_s = \frac{s_x - s_y}{2} \sin 2\phi + s_{xy} \cos 2\phi \quad \ldots\ldots\ldots\ldots (4)$$

Equations 3 and 4 now permit calculation of combined normal and shear stresses on any plane whose normal is inclined at an angle ϕ with the X-axis. The results are expressed in terms of the orthogonal components.

Principal Stresses: Equation 3 expresses s_n in terms of various values of ϕ. It is desirable to investigate the values of ϕ for which s_n will have maximum and minimum values. By differentiating Equation 3 with respect to ϕ and equating the result to zero

$$\tan 2\phi = -\frac{2 s_{xy}}{s_x - s_y} \quad \ldots\ldots\ldots\ldots (5)$$

Equation 5 defines planes of maximum and minimum normal stress. Two values of 2ϕ between 0 and 360 degrees, differing by 180 degrees, possess equal values of the tangent. Hence the two values of ϕ differ by 90 degrees. The planes thus described are called planes of principal stress.

By substituting appropriate functions of the angles defined by Equation 5 into Equation 3, the maximum and minimum normal stresses may be obtained. They are known as principal stresses. The following equation results.

$$(s_n)_{max/min} = \frac{s_x + s_y}{2} \pm \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + s_{xy}^2} \quad \ldots (6)$$

It is important to note that two values of s_n may

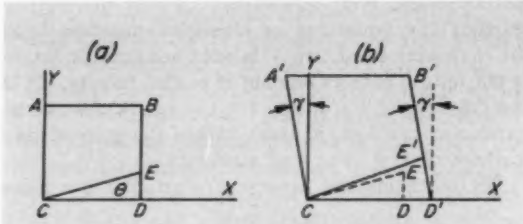


Fig. 7—Elemental cube, unstrained at *a*, strained at *b*

be obtained from Equation 6 and that two values of ϕ, differing by 90 degrees, may be obtained from Equation 5. In order to determine upon which particular plane a value of s_n calculated in Equation 6 will act, it is simply necessary to substitute one of the values of 2ϕ from Equation 5 into the general Equation 3, paying strict attention to the algebraic signs of the stresses and trigonometric functions and calculating the associated stress. The other stress value pertains to the other value of 2ϕ.

It is interesting to ascertain the values of the shear stresses on the planes of principal stress. This may be done readily by substituting values of the angles 2ϕ obtained in Equation 5 into the general equation for shear stress in Equation 4. The resulting shear stress is found to be zero. This leads to the important conclusion that the shear stresses are zero on the planes of principal stress.

Maximum Shear Stress: The planes of maximum shear stress may be located by differentiating Equation 4 with respect to ϕ and equating to zero. The planes of maximum shear will then be defined by

$$\tan 2\phi = \frac{s_x - s_y}{2 s_{xy}} \quad \ldots\ldots\ldots\ldots (7)$$

Here again the values of ϕ are 90 degrees apart, indicating that planes of maximum shear stress are 90 degrees apart. Since Equation 7 is the negative reciprocal of Equation 5, the values of 2ϕ so defined differ by 90 degrees. Therefore planes of maximum shear stress make angles of 45 degrees with the planes of principal stress.

If values from Equation 7 are substituted into Equation 4, the maximum and minimum shear stresses are

$$(s_s)_{max/min} = \pm \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + s_{xy}^2} \quad \ldots\ldots (8)$$

Significance of Principal Stresses: Not only do the principal stresses represent the maximum and minimum stress values at a point; of equal importance is that knowledge of their magnitude and direction provides a complete specification of the stress condition at a point.

Let U, V, and W represent a set of co-ordinate axes in the direction of the principal stresses. The principal stresses are s_u and s_v with $s_w = 0$ for biaxial stresses. Then $s_{uv} = 0$ since s_u and s_v are principal stresses. The requirement is to determine s_n, the normal stress on a plane perpendicular to the UV plane and whose normal makes an angle θ with the UW plane. Substituting in Equation 1

$$s_n = s_u \cos^2 \theta + s_v \sin^2 \theta \quad \ldots\ldots\ldots\ldots (9)$$

or in Equation 3

$$s_n = \frac{s_u + s_v}{2} + \frac{s_u - s_v}{2} \cos 2\theta \qquad (10)$$

To get the shearing stresses on the same plane, use is made of Equation 2

$$s_s = (s_u - s_v) \sin\theta \cos\theta \qquad (11)$$

or Equation 4

$$s_s = \frac{s_u - s_v}{2} \sin 2\theta \qquad (12)$$

Thus the principal stresses have provided a means of specifying normal or shear stresses on arbitrarily designated planes at the point.

A knowledge of the principal stresses permits a simple evaluation of the maximum shear stress. From Equations 6 and 8 it is noted that

$$(s_s)_{max} = \frac{(s_n)_{max} - (s_n)_{min}}{2} \qquad (13)$$

The designer is concerned with maximum values of normal and shear stresses when he assesses the ability of various materials to withstand them.

Graphical Determination—Mohr's Circle: A convenient graphical method is available for the solution of two-dimensional combined stress problems. It is the widely used Mohr's circle of stress.

Co-ordinate axes are first established. Normal stresses are plotted along the abscissa with tension stresses (positive) plotted to the right of the origin and compression stresses (negative) to the left. Shear stresses are plotted along the ordinate—positive above the origin, negative below. Normal and shear stresses acting on opposite faces of the element become co-ordinates of a point on the s_n s_s-plane.

The stresses acting on the right-hand and left-hand edges of the element in *Fig.* 5 consist of a positive normal stress and a positive shear stress indicated by the clockwise direction of the shearing couple. They may be plotted as point *A* in *Fig.* 6. Plotting stresses s_y and s_{yx} applying to the upper and lower edges of the element locates point *B*. Here s_{yx} is negative since it produces a tendency for counterclockwise rotation. The Mohr's circle is drawn on diameter *AB*. For convenience, radii *AC* and *BC* are labeled X-axis and Y-axis, respectively.

Examination of the circle reveals interesting relationships. By inspection

$$CA = \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + s_{xy}^2} \qquad (14)$$

which equals the maximum shear stress as computed in Equation 8.

Further inspection of the Mohr's circle reveals

$$OF = OC + CA = \frac{s_x + s_y}{2} + \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + s_{xy}^2} \qquad (15)$$

$$OG = OC - CA = \frac{s_x + s_y}{2} - \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + s_{xy}^2} \qquad (16)$$

$$\tan 2\phi = -\frac{2 s_{xy}}{s_x - s_y} \qquad (17)$$

Clearly then, *OF* and *OG* represent the maximum and minimum values of normal stress computed in Equation 6 and are therefore identified as the principal stresses. Furthermore, the value of 2ϕ is identical with the value of Equation 5.

The potentialities of Mohr's circle may be summarized as follows. From a knowledge of normal and shear stresses acting in orthogonal directions on an element subjected to plane stress, a circle may be constructed. Examination of this circle provides the magnitudes of the maximum and minimum principal stresses and maximum shear stress. The planes of principal stress and maximum shear stress may be located with reference to the *X*-axis as labeled on the Mohr's circle. Angles measured counterclockwise from the *X*-axis are positive. All angles measured on the Mohr circle are double the values of angles as measured on the body. From the circle it is also possible to determine the stresses on any plane whose normal is inclined at a specified angle ϕ with the *X*-axis. It is simply necessary to measure 2ϕ in the specified direction from the *X*-axis as labeled on the circle and to draw a radius. The co-ordinates of the resulting point on the circumference are the desired stresses.

The graphical solution is rapid and permits ready determination of the stress for any given direction. The circle assists greatly in the visualization of the state of stress.

General Case of Plane Strain: When all the strains existing on a small element of a stressed body are parallel to one plane, the element is said to be in a state of two-dimensional or plane strain. This situation exists on the surface of machine parts and is frequently subjected to investigation with strain gages.

In *Fig.* 7 the square *ABCD* represents the outer surface of an elemental cube of material in the unloaded condition. When subjected to plane strain the square becomes distorted to the shape *A'B'CD'*. It is desired to make a determination of the strain in a direction *CE* or *CE'* at an angle θ with the *X*-axis. For practical purposes, within the elastic limit the directions of *CE* and *CE'* are the same.

For typical engineering materials where displacements within the elastic limit are small compared to the dimensions of the body, strain is defined as the displacement divided by one of the original body dimensions. Thus the normal strain in a given direction equals the displacement in that direction divided by the length over which it occurs measured in the same

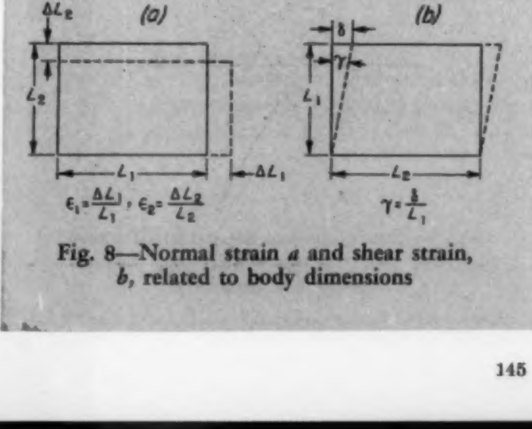


Fig. 8—Normal strain *a* and shear strain, *b*, related to body dimensions

direction. The shearing strain is determined by dividing displacement in a given direction by a length normal to the direction of displacement. This is then really the tangent of the shear strain angle, *Fig.* 8.

If the known normal strains are ε_x and ε_y and the associated shear strain is γ $(= \gamma_{xy})$, the following relations exist between the strained and unstrained conditions on the surface of the elemental cube.

Unstrained	Strained
$CE = L$	$CE' = L(1 + \varepsilon)$
$CD = L \cos \theta$	$CD' = L(\cos \theta)(1 + \varepsilon_x)$
$DE = L \sin \theta$	$D'E' = L(\sin \theta)(1 + \varepsilon_y)$
Angle $CDE = \pi/2$ radians	Angle $CD'E' = (\pi/2) - \gamma$

From these expressions and *Fig.* 7, the following equation can be developed:

$$\varepsilon = \varepsilon_x \cos^2 \theta + \varepsilon_y \sin^2 \theta - \gamma_{xy} \sin \theta \cos \theta \quad \ldots\ldots (18)$$

Of considerable value, Equation 18 provides a means of determining the state of biaxial strain at a point from a knowledge of any three strains, ε_1, ε_2, and ε_3 together with their associated directions θ_1, θ_2, θ_3, measured with respect to a set of co-ordinate axes.

Principal Strains: The directions of the maximum and minimum normal strains, hereafter called the principal strains, may be obtained by differentiating Equation 18 with respect to θ and equating to zero. Then

$$\tan 2\theta = -\frac{\gamma_{xy}}{\varepsilon_x - \varepsilon_y} \quad \ldots\ldots (19)$$

From this,

$$(\varepsilon)_{max/min} = \varepsilon_{u,v} = \frac{\varepsilon_x + \varepsilon_y}{2} \pm \frac{\sqrt{(\varepsilon_x - \varepsilon_y)^2 - \gamma_{xy}^2}}{2} \quad (20)$$

Determining Principal Strains: It is clear that the solution of Equations 19 and 20 requires a knowledge of two normal strains in orthogonal directions and the associated shearing strain. The principal strains may also be evaluated in terms of three normal strains. Such a procedure has significance from the standpoint of experimental techniques and merits examination here.

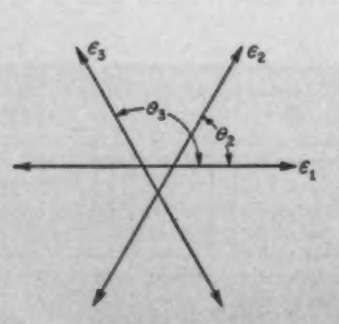


Fig. 9—Strain rosette, the pattern formed by the lines along which the normal strains are measured

Designate the three known normal strains at a point as ε_1, ε_2, and ε_3 and the respective directions θ_1, θ_2, and θ_3 with respect to the *X*-axis. For convenience let θ_1 be taken in the *X*-direction so that $\theta_1 = 0$. *Fig.* 9 shows the arrangement. With $\varepsilon_1 = \varepsilon_x$, equations may be written for ε_2 and ε_3, utilizing Equation 18:

$$\varepsilon_2 = \varepsilon_x \cos^2 \theta_2 + \varepsilon_y \sin^2 \theta_2 - \gamma_{xy} \sin \theta_2 \cos \theta_2 \quad \ldots\ldots (21)$$

$$\varepsilon_3 = \varepsilon_x \cos^2 \theta_3 + \varepsilon_y \sin^2 \theta_3 - \gamma_{xy} \sin \theta_3 \cos \theta_3 \quad \ldots\ldots (22)$$

These two equations involve only two unknowns, ε_y and γ_{xy}. The solution of the equations simultaneously provides the remainder of the data needed to compute the principal strains by the use of Equation 20.

The pattern formed by the lines along which the normal strains are measured to determine the state of stress at a point is known as a strain rosette.

Stress-Strain Relationships: Up to this point, plane stress and plane strain have been treated separately. No reference has been made to any material properties and therefore the analyses hold for all materials.

Strains are measurable quantities and therefore experimentally accessible. On the other hand, designers have traditionally dealt with stresses. Various codes specify stresses for particular applications. Furthermore, limiting properties of materials such as yield point, ultimate strength, etc., are expressed in terms of stress. Since stresses cannot be measured directly, it follows that when experimental procedures must be used, the stress distribution is obtained indirectly through an observation of strains coupled with a knowledge of stress-strain relations. These relationships are established through laboratory tests.

The complexity of the stress-strain relationship depends upon the material. For example, aelotropic materials have elastic properties which change with direction. In this case the principal axes of stress and strain do not always coincide and the relationship may be awkward. Certain materials such as rolled thin sheets are orthotropic. Here again the stress-strain relationships are complicated, although sometimes capable of analysis.

The most widely-used stress-strain relationships assume the following conditions:

1. Isotropy
2. Homogeneity
3. Perfect elasticity
4. Linear stress-strain relations

For isotropic materials the directions of the principal stresses and principal strains are identical.

To develop stress-strain relationships for an isotropic material meeting the foregoing conditions, use is made of one value for the modulus of elasticity, E, and one value for Poisson's ratio, μ. Consider such a material subjected to uniaxial stress s_x. The strains along the three co-ordinate axes are then

$$\left.\begin{aligned} \varepsilon_x &= \frac{s_x}{E} \\ \varepsilon_y &= -\mu\frac{s_x}{E} \\ \varepsilon_z &= -\mu\frac{s_x}{E} \end{aligned}\right\} \quad \ldots\ldots (23)$$

Using the principle of superposition, resultant strains may be obtained for systems in which there are also

stresses in the Y and Z-directions. Then

$$\left.\begin{aligned} E\,\varepsilon_x &= s_x - \mu\,(s_y + s_z) \\ E\,\varepsilon_y &= s_y - \mu\,(s_x + s_z) \\ E\,\varepsilon_z &= s_z - \mu\,(s_x + s_y) \end{aligned}\right\} \qquad (24)$$

These equations may be written in terms of principal stresses, s_u, s_v, and s_w and principal strains ε_u ε_v and ε_w Writing the equations for the common biaxial stress case where $s_w = 0$,

$$\left.\begin{aligned} E\,\varepsilon_u &= s_u - \mu\,s_v \\ E\,\varepsilon_v &= s_v - \mu\,s_u \\ E\,\varepsilon_w &= -\,\mu\,(s_u + s_v) \end{aligned}\right\} \qquad (25)$$

Or in terms of stresses

$$\left.\begin{aligned} s_u &= \frac{E}{1-\mu^2}\,(\varepsilon_u + \mu\,\varepsilon_v) \\ s_v &= \frac{E}{1-\mu^2}\,(\varepsilon_v + \mu\,\varepsilon_u) \\ s_w &= 0 \end{aligned}\right\} \qquad (26)$$

Equation 26 now furnishes a means of computing the principal stresses from the principal strains for the important biaxial stress case so widely encountered in design. The previous section demonstrated how the principal strains could be determined from measurements of three normal strains, utilizing Equations 20, 21, and 22.

For isotropic materials, one constant—the modulus of elasticity in shear, G—relates shear strain and shear stresses:

$$s_{xy} = G\,\gamma_{xy} \qquad (27)$$

Another useful expression relates Poisson's ratio, μ, with the moduli of elasticity, E and G:

$$E = 2\,(1 + \mu)\,G \qquad (28)$$

Any two of the elastic constants therefore specify the value of the third.

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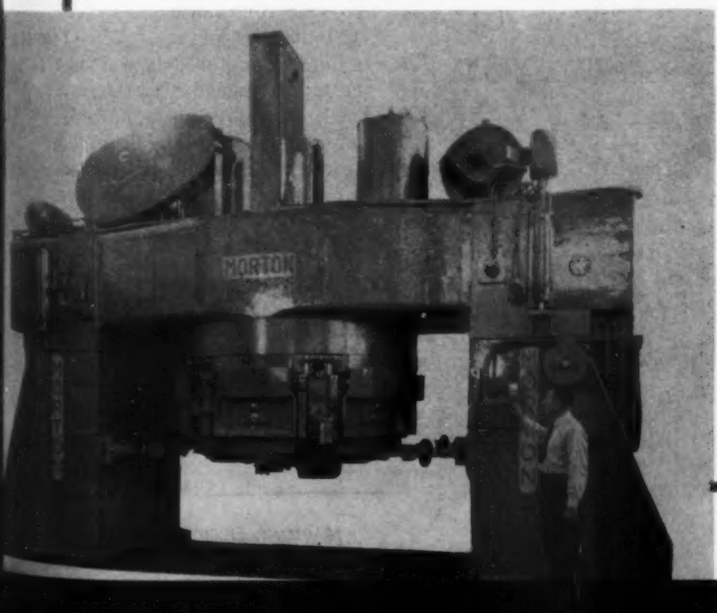
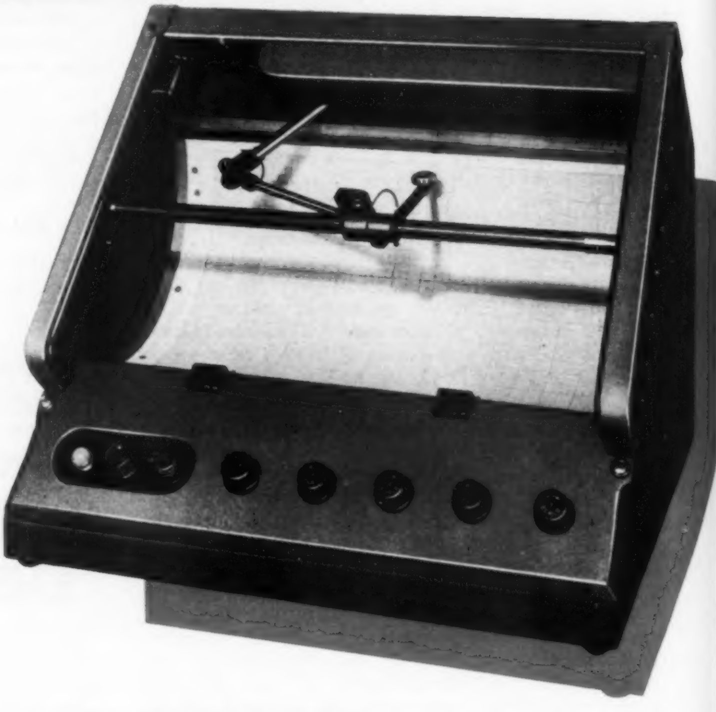
CONTEMPORARY DESIGN

Bridge Type Boring Machine

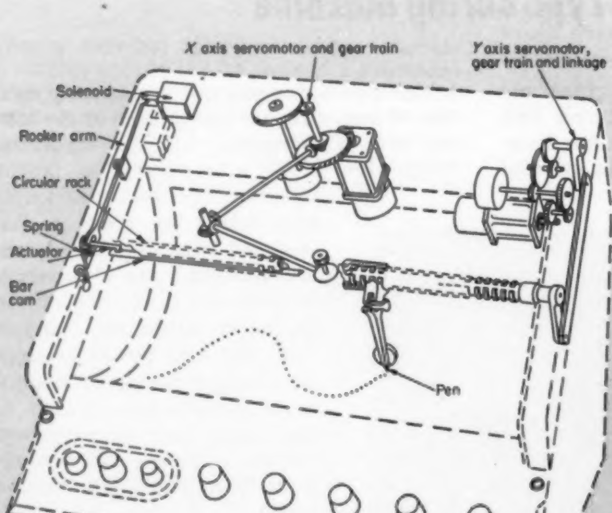
DESIGNED for heavy duty roughing and finishing operations on the turret ring bore of an Army tank hull and the ring bearing on the bottom of the turret, this inverted bridge type boring machine is extremely rigid. The 96-inch diameter boring head of the Morton Mfg. Co. machine and its supporting bridge are lifted hydraulically to provide clearance for part positioning, after which the bridge is lowered and locked to the columns. Six tool slides are incorporated in the boring head, each with independent adjustable hydraulic feed, and each can be arranged for boring or facing, or mounted at an angle for taper boring. All tool slides are mutually supporting, making a rigid arrangement. Drive is a 100-horsepower motor.



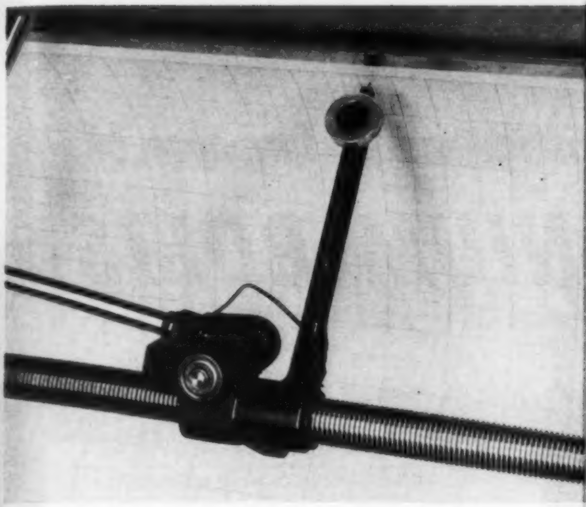
Plotter-Recorder Incorporates Unique Pen-Positioning Mechanism



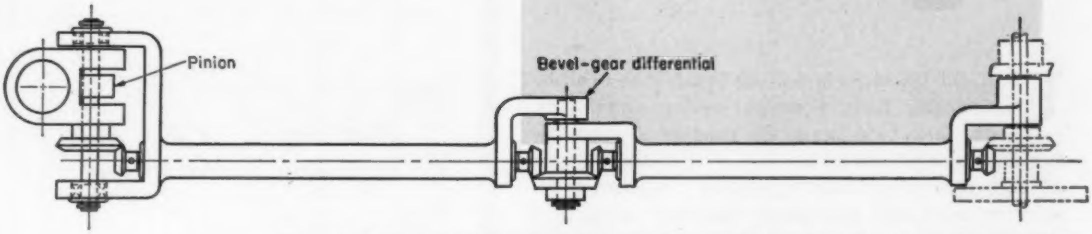
CONTINUOUS curves or discrete points are plotted on the new X-Y Plotter and Recorder developed by Librascope Inc. Incorporating an unusual pen-positioning mechanism and method of ink-flow regulation, the two co-ordinate recording instrument can be operated from analog, digital or polar co-ordinate inputs, punched cards, or a decimal keyboard unit for manual point plotting. Negative or positive values can be accepted on both axes, and the origin may be shifted to any point on 11 by 16½ inch or 8½ by 11 inch graph paper. Zero offset and scale expansion of both axes is continuously variable over a 10 to 1 range. Accuracy is within ±0.1 per cent full scale (static).



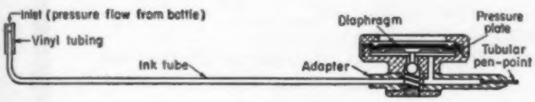
A gear train and parallelogram linkage control the *Y* co-ordinate motion through 120 degrees of arc. The *Y* axis servomotor drives a bell crank at the end of a circular rack through a push-rod linkage. The pen carriage is keyed to the rack so that pen movement is controlled along the *Y* co-ordinate but free motion along the *X* co-ordinate is permitted.

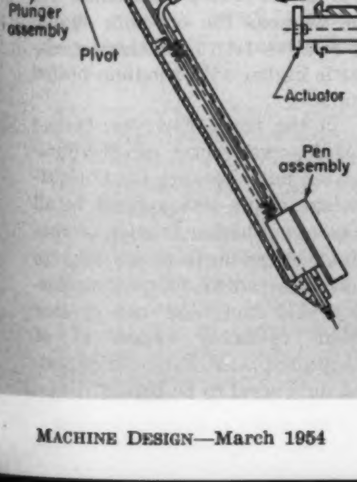


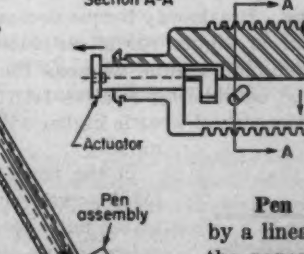
Translation of the pen carriage along the X axis is accomplished with a floating gear train. The pen assembly is driven along the circular rack by a spur pinion, in turn driven by a bevel gear train. The elbow of the floating train is a bevel-gear differential which adds an angular rotation to the pinion shaft, thus compensating for the angular difference between the two linkage arms. By this means, the pinion at the rack is "corrected" with reference to the motor gear, permitting translation of the pen carriage as a linear function of the rotation of the X axis servomotor.

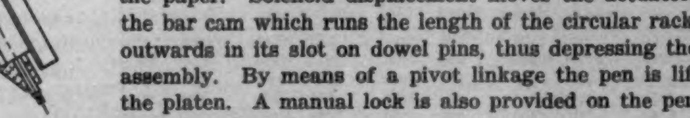


Even ink flow to the pen point despite variations in elevation of the point is accomplished with a diaphragm-controlled reservoir. The ink tube is connected to the stationary supply bottle by flexible vinyl tubing; changes of approximately 8 inches could thus produce head-pressure changes varying from 2 to 10 inches. Flooding of the pen point is prevented by a ball check valve. Depression of the diaphragm, pressure plate and pin upsets the ball, allowing ink under pressure to fill the reservoir beneath the pressure plate. As the pen writes and ink is removed, atmospheric pressure again causes the diaphragm to actuate the check valve and refill the reservoir. The diaphragm is exposed, providing a convenient means of priming the pen should the point become clogged.









Pen actuation for discrete point plotting is accomplished by a linear solenoid. Normally, the pen is spring-loaded against the paper. Solenoid displacement moves the actuator, causing the bar cam which runs the length of the circular rack to move outwards in its slot on dowel pins, thus depressing the plunger assembly. By means of a pivot linkage the pen is lifted from the platen. A manual lock is also provided on the pen plunger to retract the pen.

characteristics and performance of TITANIUM BOLTS

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USE OF titanium in bolting applications can be desirable from a weight-saving standpoint, and has been seriously considered since titanium was first available. Reported disadvantages of high relaxation under operating loads and tendency to seize and gall other materials have been definite deterrents in its adoption.

Recent tests, however, reveal that negligible relaxation occurs at normal bolt operating loads, but relaxation at high loads near the yield strength is appreciable. Galling tests indicate that combinations of RC130B titanium bolts with rolled or ground threads, together with silver or rhodium-plated stainless-steel locknuts, operate satisfactorily for a normal service life. Silver-plated nuts in combination with rolled threads appear to be slightly superior to other combinations.

Tensile strength of the 7/16-inch diameter, 20 thread-per-inch bolts was found to be comparable to the original material at room temperature and 500 F. Strength of the rolled threads seems to be slightly superior to ground threads, although the variation is small and might have been equalized with more extensive testing. Friction between titanium and rhodium-plated nuts is much higher than between silver-plated nuts and titanium; tests show that tensile load for equal tightening torque on the nut is greater with the silver-plated nuts, with and without a lubricant containing colloidal graphite. Typical test results are shown in *Fig.* 1.

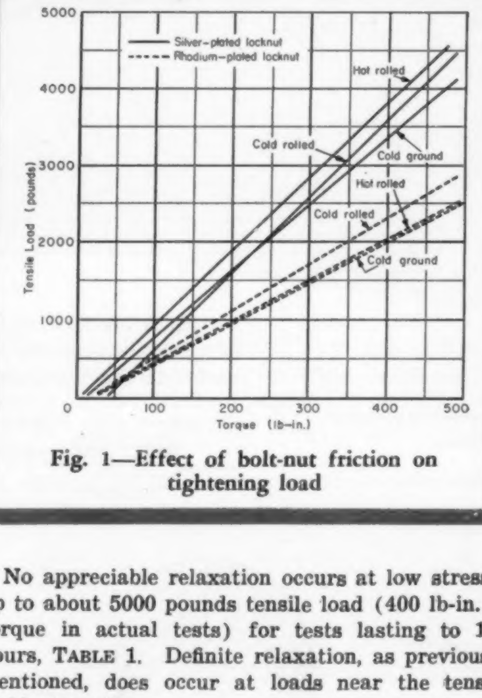


Fig. 1—Effect of bolt-nut friction on tightening load

No appreciable relaxation occurs at low stresses up to about 5000 pounds tensile load (400 lb-in. of torque in actual tests) for tests lasting to 150 hours, TABLE 1. Definite relaxation, as previously mentioned, does occur at loads near the tensile yield strength.

Silver-plated nuts have a greater tendency toward galling, as shown by tests which involved cyclic loading by tightening the locknut, heating, and unloading. Breakaway torque necessary to loosen the locknut is higher for silver-plated nuts than for rhodium-plated nuts, whereas the opposite should be true considering relative frictional characteristics. Mechanical wear is higher with rhodium-plated nuts.

In the tests, however, thread loading was lower on rhodium-plated nuts, because equal tightening torque was applied in all cases, and higher friction of rhodium-plated nuts would lead to lower thread loading. Considering this fact, and the greater wear (possibly excessive) of rhodium-plated nuts, silver-plated nuts seem to be better

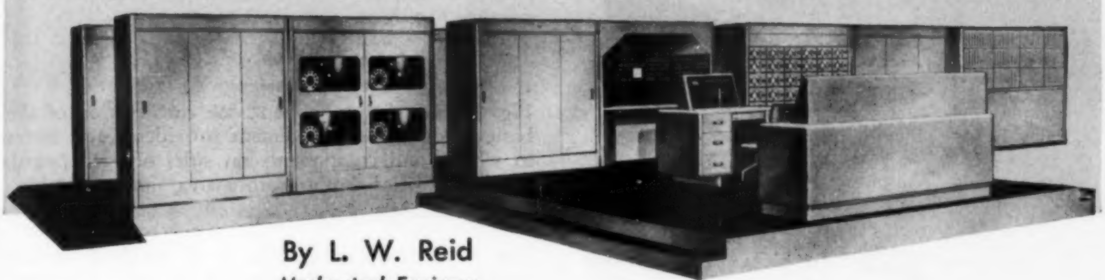
Table 1—Relaxation Tests

Thread Type	Start		1 Hour		4 Hours		24 Hours		150 Hours	
	Torque (lb-in.)	Load (lb)	Torque (lb-in.)	Load (lb)	Torque (lb-in.)	Load (lb)	Torque (lb-in.)	Load (lb)	Torque (lb-in.)	Load (lb)
Cold rolled	240	2100	240	2100	240	2100	240	2100	240	2100
Cold rolled	400	3600	400	3600	400	3600	400	3600	390	3500
Cold ground	240	2000	240	2000	240	2000	240	2000	240	2000
Cold ground	400	3370	400	3370	400	3370	390	3270	390	3270
Hot rolled	240	2100	240	2100	220	2080	220	2080	220	2060
Hot rolled	400	3800	380	3600	380	3600	380	3600	380	3600

Electronic and Mechanical

MODULAR DESIGN

. . . of complex units such as computers provides performance versatility with simplified assembly and maintenance



By L. W. Reid

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TODAY'S electronic "brains," or computers, are large, complex devices incorporating myriad vacuum tubes with their associated electronic components, various memory devices, and often complete air-conditioning systems. Despite large size and complexity, modular design using standardized components and assemblies as building blocks simplifies both electronic and mechanical design problems encountered in computers intended to perform different tasks. Application of this technique is exemplified in the computer shown in *Fig.* 1.

Basic element in the system is the unit chassis, *Figs.* 2 and 3, which is a standardized framework accommodating vacuum tubes and other electronic

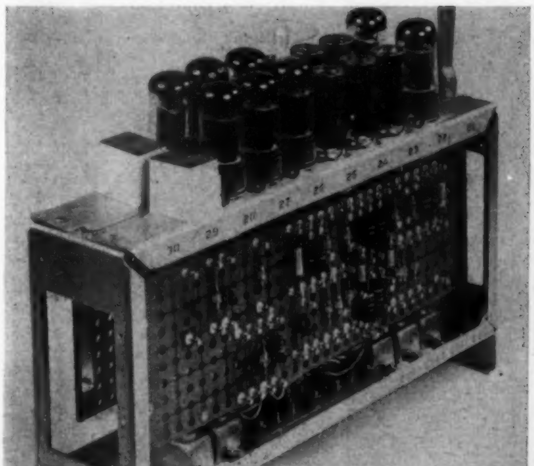


Fig. 1—Top—A complete air-conditioning system, 4500 vacuum tubes, three different types of memory elements and other equipment enable the Engineering Research Associates' 1103 computer to perform a wide range of computations. Despite its size and complexity, modular design techniques simplify both design and construction of the 1103 as well as other computers for other purposes. Ability to communicate with all known or proposed input-output devices is one of the features

Fig. 2—Basic modular-design element is a standard pluggable unit chassis. Relatively simple to manufacture, the unit may be readily removed for servicing or testing. Two nylon-base component boards within the chassis are indexed in grid fashion. A predrilled hole for a lug or for mounting an electronic component is located at each co-ordinate intersection

components plus the connecting wiring to form the computer's circuitry. Any computer system is primarily a collection of such unit chassis.

Next element in the system is the unit cabinet, any number of which can be joined to form larger cabinets, *Fig.* 4. The entire computer system is housed in the requisite number of unit cabinets. Since the desirable limit for portability is usually three unit cabinets joined, screw terminal disconnect strips are used between cabinet sections. This feature makes possible the disassembly, shipment and reassembly of a complete computer system with no unsoldering or resoldering of connections, reducing installation time to a few days.

Typical of the standardized components used is the family of magnetic memory drums. Developed and proved in computer system work, the drums provide a wide range of storage capacities and are available as stock items. Drums vary in size from a small unit for airborne use to one which has storage capacity of 1-million decimal digits. One such standard drum with the proper storage capacity is simply selected and incorporated into the system, *Fig.* 5.

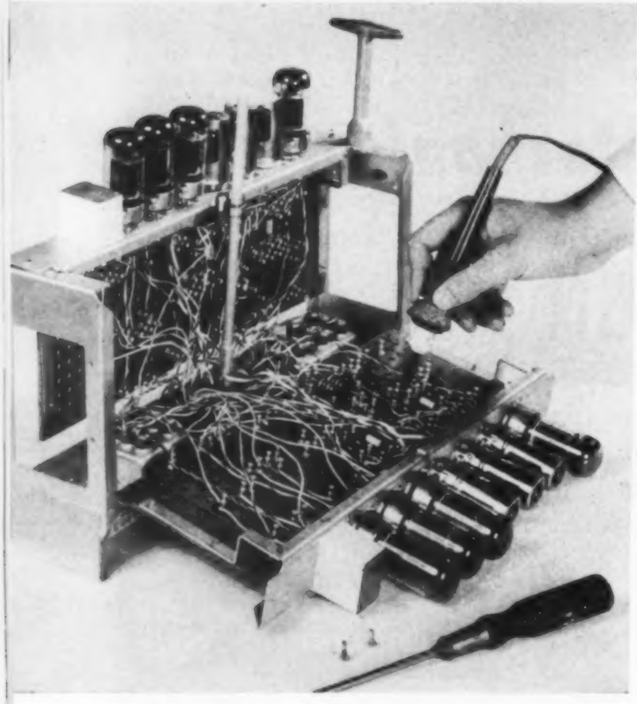


Fig. 3—Left Above—Open-book construction of the basic modular design element provides ready access to wiring and components on sides of each board, and simplifies both maintenance and assembly

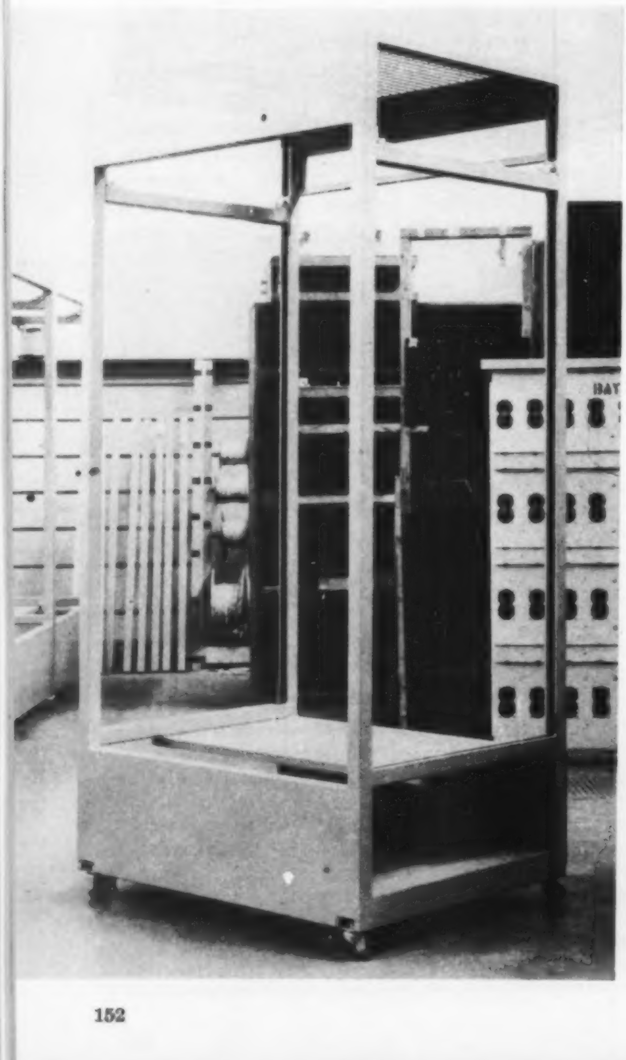
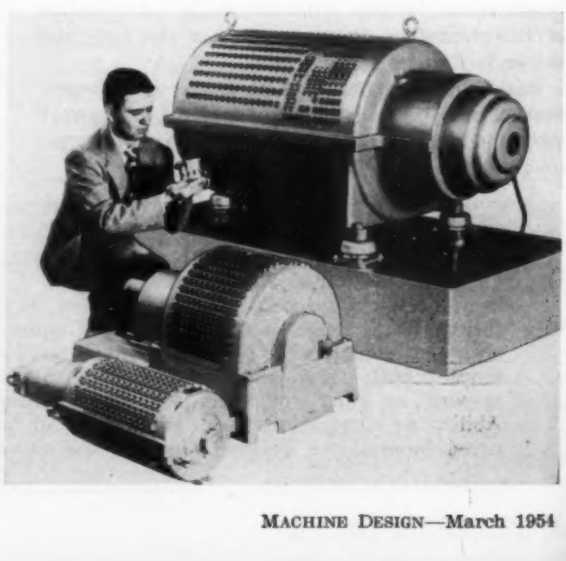
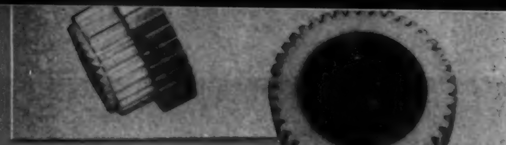
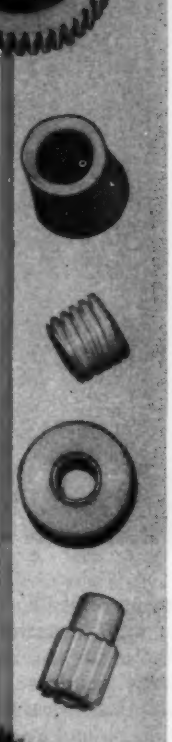


Fig. 4—Left—Unit cabinet is a standard section frame constructed primarily of welded "Unistrut" members. Back panels, separating bulkheads, and supporting frame members can be varied in size and placed most advantageously within the unit cabinet. Dimensions are 80 inches high, 40⅜ inches wide and 30 inches deep. Any number of sections can be joined to form the desired overall cabinet

Fig. 5—Below—More than 600,000 binary digits can be stored in the rapid-access magnet memory drum of the 1103 computer. Superspeed memory is provided by electrostatic storage units. Magnetic tape provides large-capacity, supplementary storage







DESIGNING

Fabricated Nylon Parts

How design flexibility plus an advantageous combination of properties can be attained by machining or stamping parts from stock nylon shapes

By R. B. Zimmerli

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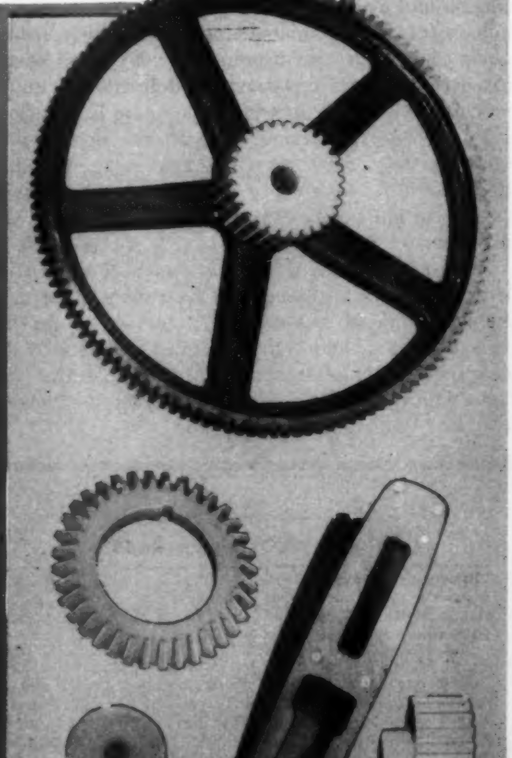


NYLON has fast become one of the most intriguing industrial materials available to design engineers—a material that is beginning to find wide application. Nylon is used in virtually every industry—in iron foundries for wear parts, in precision instrument equipment for bearings, in pulp and paper mills for corrosion resistance, in the food industry for wear parts requiring no lubrication.

What makes a material such as nylon so interesting and useful? Among its physical and mechanical characteristics, TABLE 1, nylon has several properties that are particularly useful in meeting design requirements, specifically:

1. Wear and abrasion resistance
2. Low coefficient of dry friction
3. Resilience
4. Light weight
5. Corrosion resistance
6. Ease of fabrication

Fundamentally, there are three different processes for producing nylon parts: injection molding, molding by cold pressing and sintering, and fabrication from standard shapes by standard metalworking tools. The difference between molding and machining or blanking nylon is about the same as the relationship between die casting metal parts and fabrication on turning or punch press equipment. This article discusses applications and possibilities for nylon fabricated from standard mill shapes.

Design Properties of Nylon: One application that

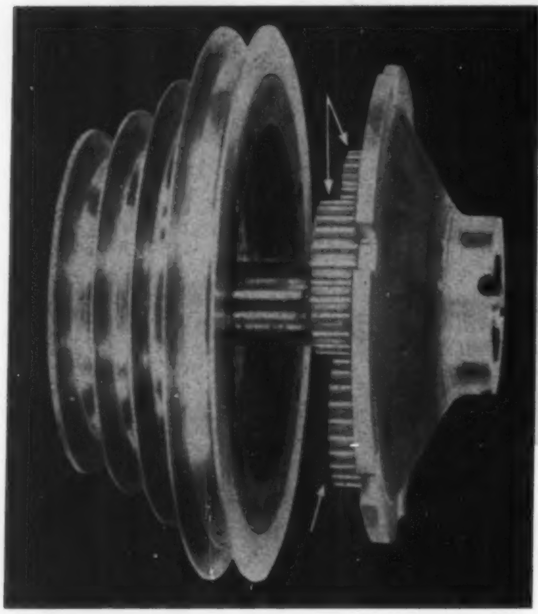


Fig. 1—Left — Resilient nylon planetary gears operate quietly and without lubrication. Driving and ring gears are both hardened steel

Fig. 2—Right — Use of nylon for steadyrest pads illustrates excellent wear characteristics of the material. Nylon jaws are said to require less frequent adjustment and to give more uniform results

graphically illustrates how a desirable combination of design characteristics can be obtained with nylon is an electric razor developed by one of the larger manufacturers. The motor shaft, with an elliptical cross-section, turns between two contacts to open and close the spark gap. The contact arms have a blanked nylon rubbing block which serves as an insulating bearing surface. Nylon is specified because of its resistance to deformation under heat, its ability to run with little or no lubrication, and its wear resistance. The shaver operates at approximately 8600 rpm. If any wear or other dimensional change were to occur on this part, the arc would not be properly interrupted and the motor would fail to operate.

RESILIENCE: One of the fundamental properties of nylon is its resiliency. This property permits the designing of gears, *Fig.* 1, and other mating parts of nylon, and the machining of parts to no-clearance or interference fits. The material will "give" and conform to rough surfaces, yet spring back to its original shape when the load is removed. This resiliency also enables nylon bearings to run with much larger clearances than those possible with metal parts. A metal bearing with large clearances will chatter and cause wear by impingement. Nylon bearings run quietly, and show little wear even without lubrication.

WEAR AND ABRASION RESISTANCE: Parts from machined nylon are being used extensively in certain wear applications. For example, several automotive manufacturers are using nylon pads for steadyrests on grinding machines and lathes. The parts are either machined from rod or from slabs, depending on the shape desired. Many materials were tested before the nylon in an effort to reduce costly downtime for maintenance. Both FM-10001 nylon and a graphite-impregnated FM-10001 formulation resisted wear longer than any other materials tested. *Fig.* 2 shows adapters built by one automotive company to hold stock size nylon rods used as steadyrest jaws in its production lathes. The adapter is designed for two purposes: (1) so that in case of wear at the surface, a feed screw at the rear can push the nylon out, and (2) to add rigidity to the nylon.

ELECTRICAL INSULATION CHARACTERISTICS: Nylon, which meets Class A temperature limits of 105 C, is an excellent general-purpose insulator. It is form-stable at temperatures as high as 240 C. Although not recommended for use at these temperatures, there is a comfortable margin of safety in heating nylon up to that point. Nylon is generally used where requirements call particularly for toughness, heat resistance or wear qualities. In this grouping are several nylon lifter buttons used in relays; nylon is used because of its excellent dielectric properties and resistance to wear and impact. These machined parts offer freedom of adjustment, and because of their wear qualities, contours remain the same over long periods.

THERMAL CONDUCTIVITY: One very important property which must be handled carefully in design is poor thermal conductivity. This property of nylon, along with high thermal expansion (about 10 times that of steel), should be considered when designing bearings or other mechanical wear parts

Table 1—Physical Properties of FM-10001 Nylon Rod*

Ultimate tensile strength (psi)	12,175
Elongation in 2 inches (per cent)	13
Hardness (Rockwell M)	84
Heat distortion temperature	
at 264 psi fiber stress (F)	182
at 66 psi fiber stress (F)	403
Impact strength, Izod (ft-lb per in. notch)	0.9
Specific gravity	1.14
Coefficient of linear expansion (per deg F)	5.7×10^{-5}
Volume resistivity (ohm-cm)	4.5×10^{13}
Power factor, 10^6 cycles	0.04
Dielectric constant, 10^6 cycles	3.4

*For ¾-inch diameter rod. Rods over 1 inch have higher hardness with considerably lower elongation and impact strength; under ¾-inch, hardness is lower, and elongation and toughness are higher.

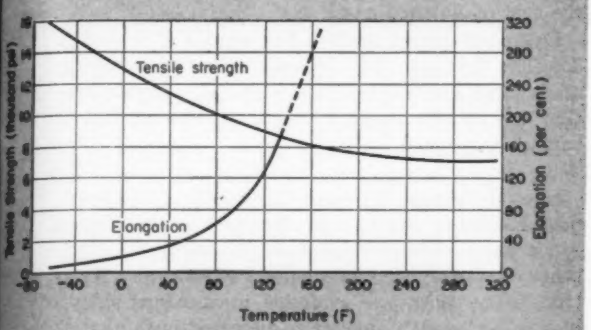


Fig. 3 — Temperature effect on tensile strength and elongation of FM-10001 nylon

subjected to high loads and speeds. Failure of such parts can be attributed to the fact that frictional heat is not completely dissipated. It is good practice, therefore, to use thin walls on sleeve bearings and, whenever possible, to leave a gap in the bearing for thermal expansion. Gear faces should be wide enough to prevent excessive tooth bending and subsequent generation of heat. Under extreme conditions, force feeding of cooling fluids will greatly extend service life of the fabricated part.

DEFORMATION UNDER LOAD: The deformation of nylon sections under load for long periods of time will be partly elastic and partly permanent. As far as known today, some of the deformation will always be elastic. Virtually all the deformation that occurs when parts are under stress at room temperature takes place during the first 48 hours of service time. Thus, valve seats and gaskets can be expected to perform satisfactorily for indefinite periods if they do not show any leaks within the first two days.

This combination of permanent and elastic deformation in nylon sections brings out engineering recommendations that, at first glance, appear to be somewhat contradictory. For example, it is poor practice to use press or shrink fits for retaining nylon parts where frictional heat can be developed. Gears and bearings generally should be pinned, keyed, cemented or snapped into position. On the other hand, nylon is useful as self-locking nuts and bolts, locknut inserts and the like where the elastic portion of long-term deformation remains effective.

MOISTURE CONTENT, CRYSTALLIZATION AND HEAT: Hardness, stiffness, impact strength and other physical properties of nylon are directly affected by moisture content and degree of crystallization, as well as by temperature. The effects of moisture content and temperature on important properties are shown in *Figs*. 3 and 4. *Fig*. 5 gives the equilibrium moisture content of FM-10001 nylon for various relative humidities. Moisture gained by rods of four different diameters soaked a varying number of hours in room-temperature water is

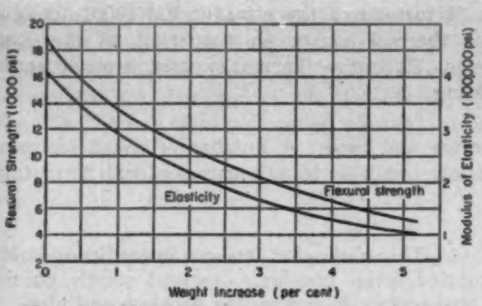


Fig. 4—Effects of moisture content (weight increase) on flexural strength and stiffness

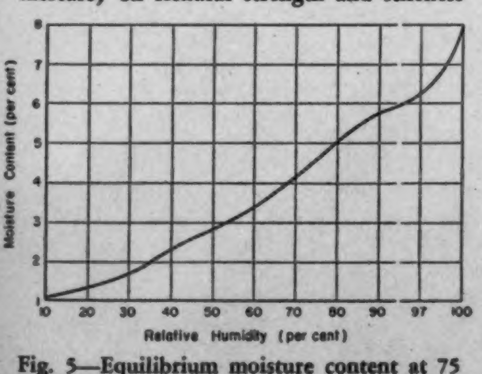


Fig. 5—Equilibrium moisture content at 75 F of FM-10001 nylon at various relative humidities

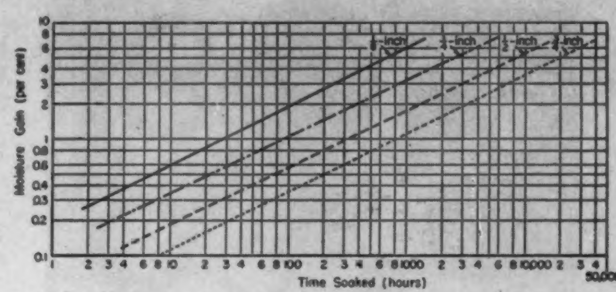
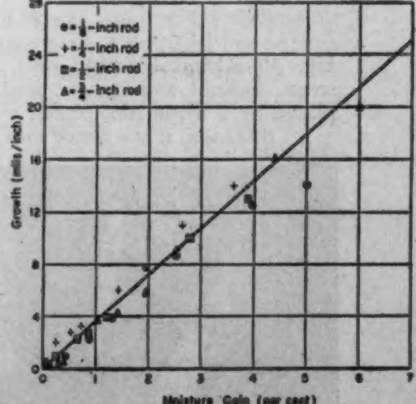


Fig. 6 — Above— Rate of moisture gain of various diameter FM-10001 nylon rods in room-temperature water

Fig. 7—Right—Expansion of FM-10001 nylon rods soaked in water at room temperature



shown in *Fig.* 6, while *Fig.* 7 charts the hygroscopic expansion of various diameter nylon rods after soaking in water at room temperature.

These effects of moisture absorption on nylon parts can be offset by conditioning the parts to equilibrium moisture content before final machining. It is advisable, however, to resort to the moisture stabilization procedure only when absolutely necessary, since this process adds substantially to the cost of finished parts.

Degree of crystallization has similar effects, *Fig.* 8. The greater the crystallization, the harder, stiffer and less ductile the material. Crystallization can be increased by heat treatment, but it cannot be reduced.

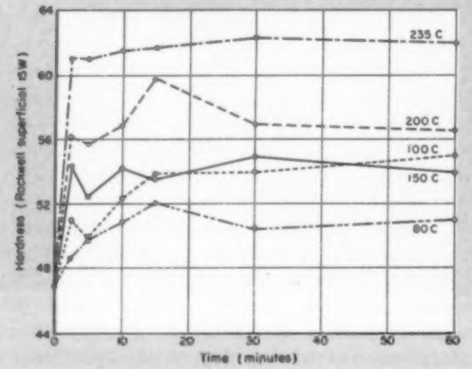


Fig. 8—Above—Time and temperature effect (crystallization) on hardness of 0.028-inch thick FM-10001 nylon strip

Fig. 9 — Below — Mechanical parts requiring no lubrication are frequently made from a graphite-filled formulation

Special-Purpose Nylons: One of the special-purpose FM-10001 nylons has already been mentioned —a graphite-filled formulation (Nylatron G). Another is a molybdenum-disulfide filled type (Nylatron GS). Both are available in standard shapes of rod, strip, slab and large diameter bushings, as well as molding powders. These formulations offer the highest tensile, compressive and flexural strength of any nylons. While harder than the unfilled FM-10001 nylon, they also provide unusual wear resistance and low coefficients of dry friction. The impact strength is somewhat lower due to the addition of the filler.

One of the most unusual properties of these filled formulations is their coefficient of thermal expansion. Both Nylatron G and GS have coefficients about half that of unfilled FM-10001 nylon. The filled formulations find a wide variety of applications in mechanical parts such as cams, rollers, cam followers, thrust bearings, bushings, scraper blades and similar components which cannot be regularly lubricated, *Fig.* 9. One particular application of Nylatron G is in conveyor chain channels for food processing plants. These channels reduce noise and wear, and decrease power requirements for operation of the conveyor. The filled formulation is used instead of the straight FM-10001 to minimize thermal expansion occurring in the long lengths. Excessive thermal expansion would cause buckling.

Forms and Sizes: A number of nylon formulations are available in mill shapes of rod, strip, tub-

Fig. 10—Original design, left, requiring a metal screw, locknut and ceramic insulator, has been replaced by a nylon pin, center, which forms its own threads as it is screwed into the nut, right

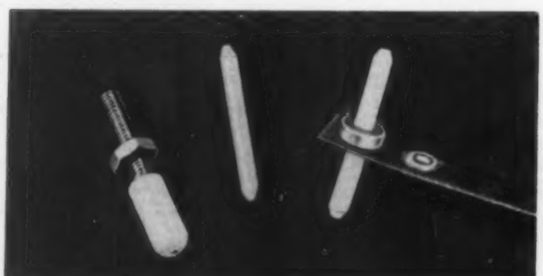
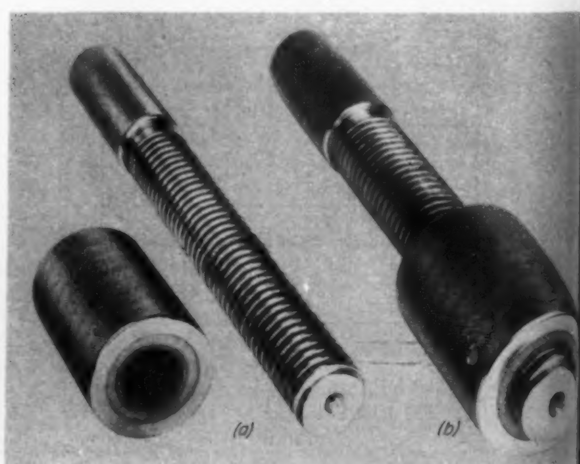


Fig. 11—Threaded nylon feed-nut insert fitting inside a knurled metal housing, *a*, screws snugly on the lead screws, *b*, eliminating backlash and play



ing, slabs and cast bushings. FM-10001 nylon, the hardest and most widely used formulation, for example, is available in all these forms. Round rod is available from 1/16 to 5 inches in diameter. Strip and slabs are stocked in thicknesses from 0.010 to 1 inch. Cast bushings can be supplied in diameters up to 10 inches for turning or milling on standard metalworking lathes. Disks range from ¼ to 2 inches thick, and tubing OD's run from 0.035 through 10 inches. From these various sizes, finished parts ranging from 0.01-gram to 21 pounds 4 ounces have been made. Many companies carry an assortment of standard shapes and sizes in their stockroom along with standard metal shapes.

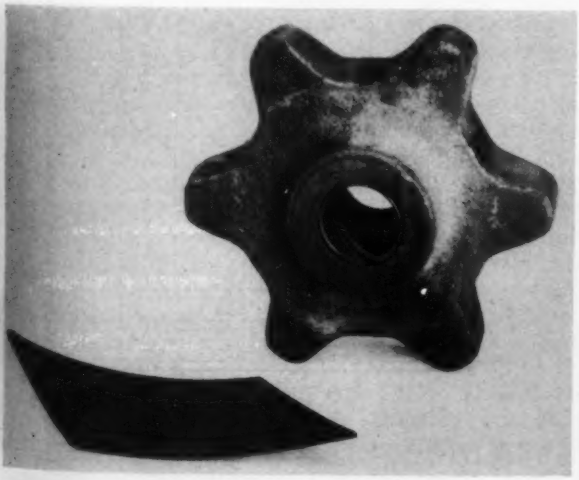
Utilizing Nylon's Properties in Design

Nylon's unique physical properties pose design considerations that must be observed if maximum performance is to be obtained. Direct substitution of nylon for other industrial materials can result in excessively high costs and poor performance if these aspects are not fully understood. Each application should be considered individually, with nylon characteristics in mind.

Elasticity and Resilience: Nylon's elasticity and resilience make possible many designs which result in cost savings. For instance, snap-fits with nylon often save assembly time and costly machining. Furthermore, it is possible to make assemblies using unthreaded nylon parts in conjunction with standard nuts or stamped metal nuts of various types. A straight nylon shaft can be screwed into a standard nut, forming its own threads as it goes, *Fig.* 10. It can also be pushed through stamped nuts with good results.

One of the newest and most intriguing applications for nylon is found in a recently designed milling machine. In this particular application, a feed nut made from nylon completely eliminates backlash in the milling machine, *Fig.* 11. The nylon nut was machined from solid bar stock and inserted into the knurled metal housing. Resiliency of the nylon causes constant pressure on the lead screw, thereby holding it firmly in place. Backlash and position changes due to vibration are completely eliminated. In addition, the threads cut into the nylon were machined very readily. No problem of wear from metal chips or loosening has been encountered during the entire test period.

Fig. 12—Graphite-filled nylon strip snaps into place to serve as a bearing surface inside this cast-iron sprocket



Nylon lock washers are finding a diversified field of applications. In various types of hardware, nylon washers are used to keep doors and panels from swinging too freely. An interference fit between the hardware and the nylon washer keeps the doors open at the position set by the opener. A similar application was found practical by a large automotive company to maintain a set position on a rear-view mirror. The use of a blanked nylon washer proved to be a very economical method of holding the mirror, even during the vibration set up in automobiles. Along similar lines, nylon is expected to gain acceptance as a material for refrigerator hinges and related applications where a friction fit is desired and smooth operation must be maintained.

Nylon's resiliency makes relatively wide tolerances and standard machined surfaces generally satisfactory. This is fortunate, since nylon's high coefficient of thermal expansion, together with the dimensional changes which occur on moisture absorption, sometimes render close machined tolerances of academic interest only. What seems most interesting in many nylon applications is that no-clearance and even interference fits of gears and bearings do not cause difficulties unless loads and speeds are great enough to cause overheating. The close tolerances customarily specified on metal mechanical parts will often prove needlessly expensive and wasteful.

Low Friction and Wear: Many newer designs for bearings have utilized nylon's excellent low friction and wear properties. These designs specify that the bearing surface be made by spiral-wrapping nylon strip to the inside of a metal housing, as shown in *Fig.* 12. Use of die-cast metal housings and thin blanked nylon liners has resulted in better performance as well as substantial economies. The nylon liners are held in the housing by slight flanges at the ends of the housing. No cement is required since the nylon is sufficiently rigid and resilient to snap into place. This type of bearing has been used in a variety of process equipment, farm equipment, and even in juvenile vehicles.

At the present time, the largest number of blanked nylon applications are in thrust bearings. A number of fractional horsepower motors use ny-

lon thrust bearings blanked from continuous coils of strip. Although many thrust washers are blanked from straight FM-10001, some users have found that the graphite-filled formulation has better wear and frictional characteristics.

Toughness: While nylon exhibits outstanding wear and frictional properties even under adverse lubricating conditions, parts made from nylon show other interesting design features. The toughness of blanked washers enables the design engineer to use thinner sections. In fact, because of the abrasion resistance characteristics of nylon it is usually practical to reduce the thickness of the nylon washer considerably over the former materials used. The great majority of thrust bearings blanked from strip are below 1/16-inch in thickness. Many are as thin as 0.012-inch or 0.020-inch. While other bearing materials at this thickness are not practical from a breakage or performance standpoint, nylon performs with full efficiency.

Temperature and Corrosion Resistance: Blanked nylon oil seals have the thermal and chemical resistance, and the wear and toughness properties needed for applications involving reuse of the seals. For example, several ball-bearing companies have adopted thin nylon seals blanked from continuous coils of strip. One automotive company is using a nylon oil-seal washer in a new design for automatic steering equipment (MACHINE DESIGN, July, 1953, Page 123).

One of the earliest uses of nylon thrust bearings was in rotating lawn sprinklers. One manufacturer was plagued for several years by an abrasion problem, solved finally with a nylon part. Tap waters used in the application had eroded several different metal washers. The customer first tried washers blanked from FM-10001 nylon, which were satisfactory. Later tests of the graphite-impregnated formulation resulted in "considerably longer life and better wear qualities."

Fabricated Part Design

Supplied in standard mill sizes and shapes, nylon can be machined or blanked readily on standard metalworking equipment. Fabricating the material in this manner means lower tooling costs, greater design flexibility, closer tolerances than available before, and low-cost pilot runs.

Design for Machining: Several examples may help to point out various areas in which machined nylon parts can be used effectively. Designed for use on electrical equipment, the small nylon parts shown in *Fig.* 13 were all machined on automatic screw machines. Note that the parts have been simplified so that the pieces can be easily turned on standard automatic metalworking equipment without a secondary operation. Because of their small size, the parts were more economical to machine from stock shapes than to mold, even though the quantity of pieces was large.

Several electrical equipment manufacturers are using nylon insulating screws made on screw machines for holding relay stacks together. The use of a nylon screw makes the usual insulating tubing between the metal screw and the electrical contact points unnecessary. Machining the screw to an interference fit also eliminates the need for lockwashers.

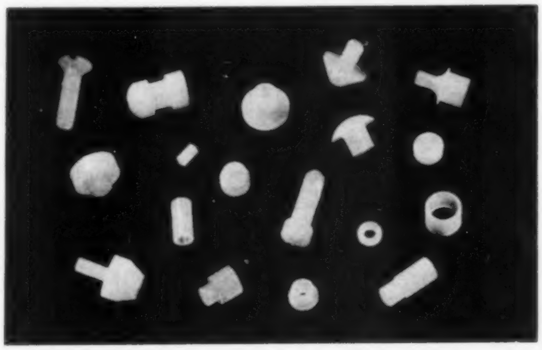
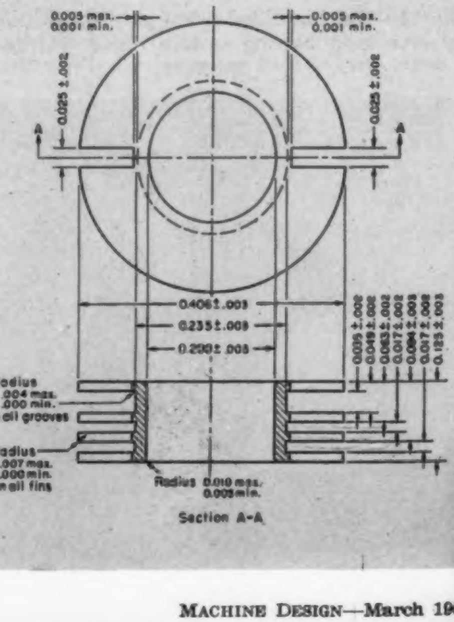


Fig. 13—Above — Tiny nylon parts for electrical equipment, easily machined on automatic screw machines

Fig. 14—Nylon bobin for electrical units, above, is machined at high production rates. Note extremely thin machined sections, below



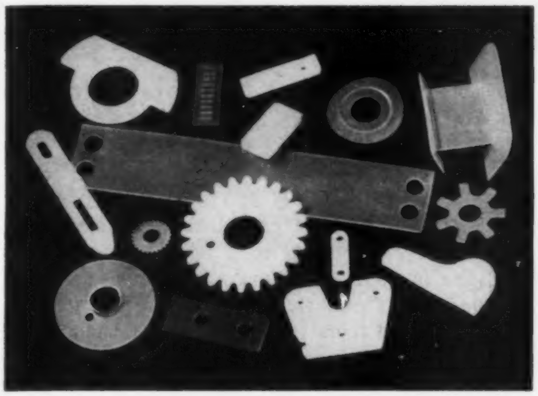


Fig. 15—Irregular shapes from nylon can easily be blanked on standard punch presses

The trend to smaller, more compact parts has aided this development of nylon for electrical applications. Since nylon is tough even in very thin cross-sections, miniature coil forms can be machined very readily. Many other insulating materials have to be made in heavier wall sections to prevent damage by breaking or chipping, either in assembly or in actual operation.

A drawing and picture of one small-size production bobbin are shown in *Fig.* 14. The grooves of the bobbin can be turned on automatic screw equipment. However, a secondary operation must be performed by milling the side slots. Machining would not be economically practical if the print called for a rectangular hole or rectangular periphery. Being round on both the OD and ID, the part is competitive with molding since it was machined at high rates of production on automatic turning equipment. Note the thin walls of the flanges.

Like most materials, nylon's impact strength is greatly affected by stress concentration. Therefore, sharp corners should be avoided and generous fillets supplied where possible. Whitworth threads, with their rounded roots, have been shown to have ten times the impact strength of American Standard threads.

Design for Stamping: Although the production of nylon parts by blanking parts on punch presses is still in its infancy, this method is growing rapidly as more engineers become aware of the practical and economical aspects of blanking the material.

Nylon parts are blanked from continuous coils of strip on standard punch press equipment. The same stock-feeding methods used for metals can be applied. For best results, it is usually advisable to punch nylon at high speeds. Many nylon parts are punched at speeds up to 600 pieces per minute. These higher speeds have an additional advantage of lowering the unit cost of the blanked nylon parts.

The shape of a part does not necessarily rule out the blanking process. Irregular flat shapes have been blanked out of nylon as shown in *Fig.* 15. The sizes of the parts are limited to the availability of strip—4 inches is the widest currently available. On larger diameter parts, factors such as dimensional changes caused by thermal or hygroscopic expansion have to be considered before drawing up final tolerances. These characteristics make nylon quite different from metals from the design standpoint.

Because nylon is not as stiff as metals, flanges should be kept at least 1/16-inch wide. On nylon material thicker than 1/16-inch, the flanges should be at least as wide as the thickness. Clean edges can be stamped on material up to 1/16-inch thick. Above that thickness, the edges become rougher and the depth of valley at center of thickness increases. This concave edge is due to the compressibility of nylon and in most cases does not detract from the performance of the part. A recent discovery proved that a satisfactory deburring operation can be performed as a secondary operation. This operation eliminates all feather edges and is relatively inexpensive for large runs.

Applications for blanked nylon parts include blanked gears, oil seals, lock washers and dust guards, blanked or drawn insulating shells, wear strips for sliding surfaces, etc. Blanked gears are used in small timing mechanisms, lawn sprinklers and a variety of other applications.

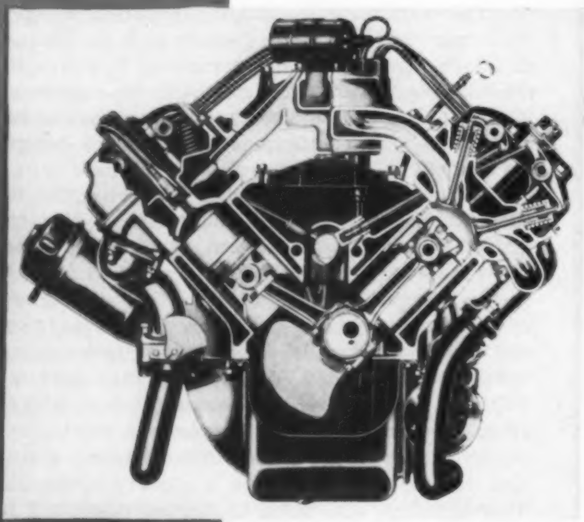
Complex shapes can also be cold or hot-formed from nylon strip or rod. Thus, small rivets are punched and cold-headed from strip in a single operation. Dust seals, coil forms and channel or angle shapes are readily cold-stamped or rolled to shape. It is practicable to cold or hot-head nylon rivets at assembly.

Conclusion: The potential applications for fabricated nylon are far from realization. So far, nylon has been used chiefly to extend service life and replace other materials. Certainly this material has shown an amazing usefulness while holding almost unlimited promise for the future. Strongly in its favor is its current availability in a wide range of stock sizes and shapes. This variety of shapes is especially important from the design standpoint because it simplifies design problems and reduces fabricating costs. While the initial cost of nylon is likely to be slightly higher than that of other materials, the savings in improved performance, long equipment life and/or quicker assemblies more than offset the original difference in price.

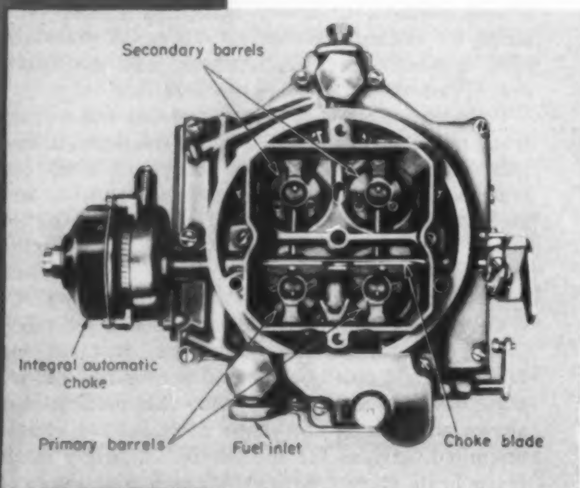
They Say . . .

"The disappearance of the gap between the research laboratory and the application of science to industry means that the successful industrial leaders of the future must understand engineering and scientific methods, preferably from training in one of these disciplines."—Henry D. Smyth, *member, United States Atomic Energy Commission.*

Redesigned Chrysler Engine Develops 235 Horsepower



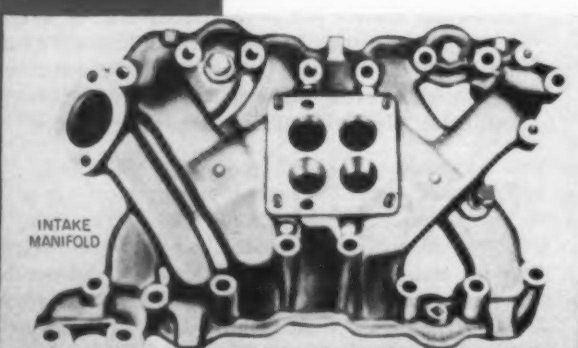
INCREASED air capacity is the secret of Chrysler's jump to a 235-horsepower engine in 1954 New Yorkers De Luxe and Imperials. The basic Fire Power V-8 engine is unchanged from the original 1951 basic model; a wide-open intake system and free-flowing exhaust system account for the increase in horsepower rating from 180 in 1953 to 235 this year. Displacement and compression ratio have not been changed, and future potentials for the engine are demonstrated by a racing version which has run up to 430 horsepower for 1000 hours without mechanical attention. The standard production model develops 235 gross horsepower at 4400 rpm; maximum gross torque is 330 lb-ft at 2600 rpm; and specific output is 0.710-bhp per cubic inch.



First major modification to increase air flow is in the carburetor. A dual-throated carburetor was used in the original engine. But for higher engine output, a four-barrel carburetor is used on the Fire Power "235". The carburetor can be considered as two dual carburetors built into a single body; the primary side has all necessary components for independent operation, while the secondary dual carburetor supplements the primary only near maximum power.

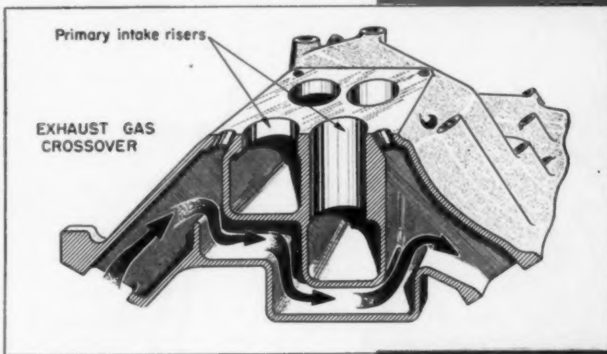
Throttle of the secondary unit remains closed until the primary throttle is about halfway open. At this point a mechanical linkage opens the secondary throttle at a rate that enables both to reach wide open position simultaneously. When the choke—located in the primary side—is in operation, a mechanical latch prevents the secondary throttle from opening, so that only the primary side is used during warm-up. After the choke is wide open, the latch is released.

Metering rods and accelerator pump are omitted in the secondary unit, air flow alone determining the amount of fuel emitted from the secondary nozzles. To prevent air leakage around the throttle valves when the secondary is not operating, the valves are offset so that manifold vacuum tends to close the throttle.

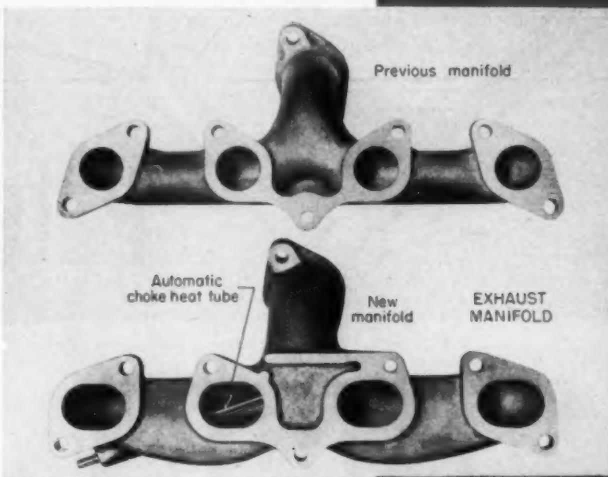


Changes in manifolding are the second modification of importance. In the intake mani-

fold, the design is aimed at the most desirable compromise between a midrange (high-torque) and high-speed (high-power) output. A secondary pair of risers and increased cross-sectional area constitute the main design changes. The manifold is divided into two isolated distribution systems, each feeding two center cylinders in one bank and the two outer cylinders in the other. Each system has two risers, one fed by a primary barrel and the other by a secondary barrel of the carburetor.

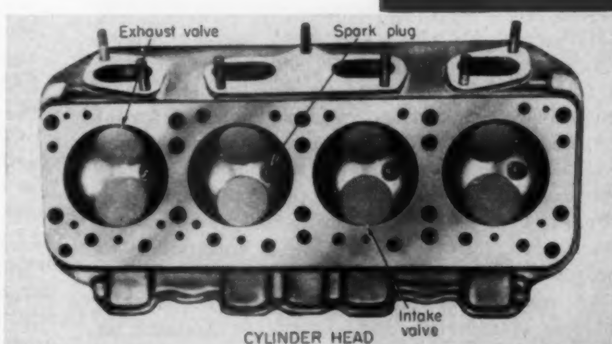


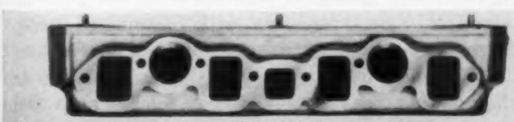
Exhaust gas crossover passage in the intake manifold has been modified to eliminate the heat tube for the automatic choke, which is now located in the right-hand exhaust manifold where it is more sensitive to operating conditions.

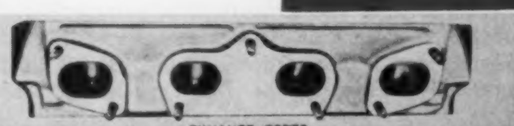


New exhaust manifold has been streamlined and considerably enlarged. Manifold outlet to the exhaust pipe has been increased from $1\frac{7}{8}$ to 2 inches in diameter, with a corresponding increase in exhaust-pipe diameter. A dual exhaust system, with separate exhaust pipe, muffler and tailpipe for each side of the engine, is used.

Cylinder head has also been redesigned for increased air capacity—the third major modification. The hemispherical chamber head with opposed lateral valves has enabled larger valves to be used without crowding, and room still remains for increases in valve diameter if desired. Intake valve diameter has been increased by $\frac{1}{8}$-inch and exhaust valve diameter by $\frac{1}{4}$-inch. Intake and exhaust ports have been enlarged to match the enlarged valves. Exhaust ports at the outside of the head have been changed from a circular to an elongated opening, almost doubling the port area, while intake port area is increased over 50 per cent.







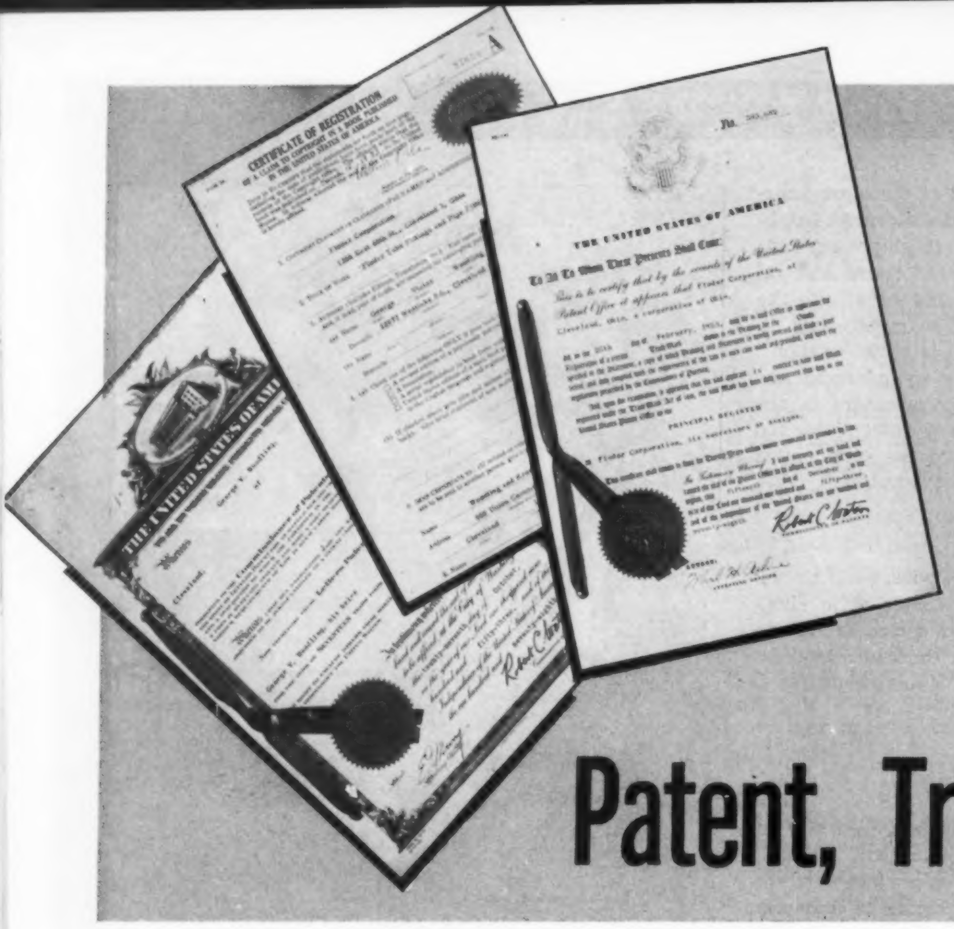


Fig. 1 — Documents certifying ownership of a patent, a trademark and a copyright

Patent, Trademark o

PROTECTION of the property rights of inventors and others who create ideas or expressions of ideas is fundamental to our industrial growth. In recognition of this fact the law has characterized patents, trademarks and copyrights, *Fig.* 1, as industrial property, which is recognized as the source of our greatest wealth. The three categories of protection cover well defined areas, but differences between them are sometimes not too clearly understood. It is the purpose of this article to explain these distinctions, to outline the differences in procedures and "life," and to show how to tell quickly which form is appropriate to the job at hand.

An outline showing the distinctions between patents, trademarks and copyrights appears as TABLE 1. Examples of the distinctions are illustrated in *Fig.* 2, which is a typical catalog covering a *patented* invention embodied in a *trademarked* product and announced in a *copyrighted* publication. The tube fittings are the subject of patents; the trademarks "Griptube" and "Fluid Fortress" distinguish the fittings manufactured and sold by the Flodar Corporation from those made and sold by other companies; the catalog with its contents (illustrations and text) is the subject of copyright.

Trademarks and Patents: Property in a trademark should not be confused with property in a patent. A company may develop a new device and obtain a patent to protect the unlawful appropriation of the construction of the device and, at the same time, may designate the new device by a trademark and obtain a registration to protect the device from being unlawfully sold by others under the same trademark. The trademark indicates the origin of goods and does not give any exclusive property right in the article to which the trademark is attached, or in any invention residing in the article. The right to exclude others from manufacturing, selling and using a device can be obtained only by complying with the patent laws. Consequently, two companies may manufacture and sell the same unpatented device under dissimilar trademarks provided the relationship between the companies presents no element of unfair competition.

The Constitution of the United States does not

This article is based on material prepared for the new edition of Mr. Woodling's book *Inventions and Their Protection.* Based on a series of articles which appeared in MACHINE DESIGN, the book has been greatly expanded and largely rewritten. Publisher of the new edition is Matthew Bender and Co., Albany, N. Y.

rk or Copyright?

By George V. Woodling

How to distinguish between the three methods of protecting ideas

authorize the protection of trademarks as it does that of patents and copyrights. A trademark is sometimes referred to as an "orphan" in the law. In a trademark, it is its adoption and use on the goods in commerce that create the rights which the courts protect. The continued use of a trademark by its owner is essential to its validity and right to exclusive ownership. A trademark is not an invention (patent), nor is it a writing (copyright), as specified by the United States Constitution.

Ownership of a trademark is a common-law property right founded principally upon the basic concept of unfair competition and the general equitable principles governing such practices. An example of unfair competition would occur when one party palmed off his goods as those of another; it does not necessarily involve a trademark, but more often it does. Throughout the course of the years, the courts have enunciated sound principles for the protection of trademarks and now many of these principles have been incorporated and codified into the new trademark law which affords more direct and effective redress against unlawful use. One of the best ways to obtain effective redress against unlawful use of a trademark is to register it in the United States Patent Office.

The Patent Office serves as a kind of clearing house for the prevention of duplication in trademarks. All registered trademarks are placed on file and open to public inspection. An interested party may, by making a search of the records, avoid the adoption of a trademark already adopted by another.

In other words, the burden is on the newcomer or prospective user or adopter of a trademark to make a trademark search to ascertain whether the proposed mark is free for use in commerce, otherwise he might be faced with a possible infringement suit if he should carelessly adopt a mark already in use.

A further advantage is that the owner of a trademark is entitled to mark his product with the notice: "Reg. U. S. Pat. Off." or the letter "R" enclosed in a circle, thus ®.

Compliance with the marking statute constitutes legal notice to the public that the owner has registered the mark and claims its rightful ownership. It is important that all such registered marks carry this notice, for the reason that the trademark laws provide that it is the duty of the registrant to give the public notice of the fact that the trademark is registered. Failure to include this notice with the trademark prevents the recovery of damages for past infringement, except on proof that the defendant was directly notified of infringement and continued the same even after such notice was given.

Fig. 2 — Catalog illustrating three categories of protection: a patented invention embodied in a trademarked product which is announced in a copyrighted publication

PROTECTING IDEAS

Copyrights and Trademarks: The subject of copyrights is directed primarily to the finer arts, which are afforded protection against copying by others. A partial list of copyrightable material may include books, periodicals, catalogs, lectures, sermons, musical compositions, works of art, models or designs for works of art, maps, photographs, commercial prints and labels, and motion pictures other than photoplays. Copyrights are not to be confused with trademarks, which merely protect an identifying mark or symbol. A person uninformed in these matters often has the wrong impression that a trademark can be protected by a copyright. This is not true as trademarks are not considered copyrightable material.

A person desiring to apply for a copyright registration may write to the Register of Copyrights, Library of Congress, Washington, D. C., for information and for the proper copyright form for the particular material for which he wishes to register a copyright. This service, including the information and the copyright form, is free of charge.

Material which is to be registered as a copyright must first be published and must bear the notice of copyright. "Copyright (year) by (name of copyright proprietor)." See the copyright notice appearing on the catalog of *Fig.* 2. Copyright may then be obtained by filing an application for registration in the Copyright Office.

There is a general belief that the copyright proprietor has no legal right to apply this notice of copyright upon his material to be registered until he has obtained the copyright registration from the Copyright Office. This assumption is incorrect. It is very essential that the notice of copyright be printed upon the material to be copyrighted as of its first publication and before mailing it to the Copyright Office. Unfortunately, when copies have once been published without the copyright notice, there is no legal way of securing a registered copyright for the work. Even the addition of a subsequent publication bearing the correct copyright notice will not cure the defect of a prior publication without such notice.

Table 1—Distinction Between Patents, Trademarks and Copyrights

	Patents	Trademarks	Copyrights
Subject Matter	Process, machine, manufacture, or composition of matter, plant and design*	Word, name, symbol or combinations thereof indicating origin of manufacture and sale	Books, paintings, photographs, sculpture, charts, drawings and commercial prints and labels
Rights Granted	To exclude others from manufacturing, using and selling devices embodying patented inventions	To keep others from using mark on similar goods which might confuse the public (prevent others from unfairly competing with similar mark)	To bar others from copying (Copyright means right to copy, but gives no right to ideas expressed in work)
Who May Obtain	Any inventor (adult, minor, citizen, alien, and any natural person; dead or insane inventor by representative)	"User" of the mark; only person or company who has actually adopted and used the mark on goods in commerce	"Author" of the material; may be an individual or corporation (Owner of the right)
Term	17 yrs not renewable*	20 yrs, renewable from term to term as long as mark is used; affidavit of continued use after five years	28 yrs; renewable for 28 yrs more—total of 56 yrs
Where to Apply for	Patent Office—Commissioner of Patents	Patent Office—Commissioner of Patents (Trademark Division)	Library of Congress—Register of Copyrights
Document	Letters Patent	Trademark Registration Certificate	Copyright Registration Certificate.
Fees	Filing: $30.00 Final: $30.00 Design Patents (see footnote)*	$25.00	$4.00—except for prints and labels. This fee is $6.00

*Design patents cover only appearance or outside effect (ornamental rather than useful) and differ from other patents in that they are issued, at the option of the inventor, for terms of 3½, 7 or 14 years, with corresponding fees of $10, $15 and $30.

OPTIMUM DESIGN

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. . . A concept for improving the balance of design factors, here applied to dynamic loading of power chains, bolts and mechanism components

DESIGN for maximum load or power capacity with minimum weight requires consideration of both static and variable stresses in the materials of construction. In many situations a comparable job could be done either by a large slow-running machine or by a small high-speed machine. Such is actually the case in engine design. In the slow-running machine static stresses are more important than alternating stresses while the reverse may be true for the high-speed machine. Somewhere between the extremes of slow and high speed there is an optimum speed at which the static and endurance properties of the materials are both developed to their fullest advantage.

In other situations prestressing can be used to increase the load capacity of a machine part by improving the stress distribution. Again, there is an optimum preload which gives the most favorable balance.

This article discusses the philosophy of optimum design and presents some specific examples to illustrate the general approach, which is applicable to a wide variety of design problems.

Usual steps in designing a machine are:

1. Determine the kinematic scheme
2. Determine the forces
3. Proportion the parts

When these steps are applied the speed usually has either been specified or assumed. The steps are not independent when inertia forces are considered,

Fig. 1—Typical periodic loading which may be considered as a steady load F_{av} with a superimposed reversed load F_r

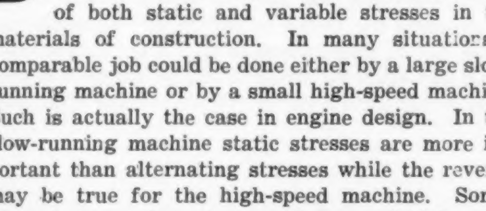
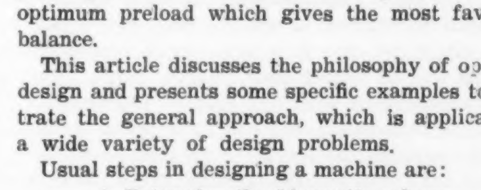


Fig. 2—Relationship between yield and endurance strengths for typical metals used in machine construction. Soderberg failure line is conservative and is consistent with the general practice of basing design on the yield rather than the ultimate strength. On the safe working stress line, F is the general case such as represented by *Fig.* 1, s_e/N is for completely reversed loading and s_y/N is or static loading



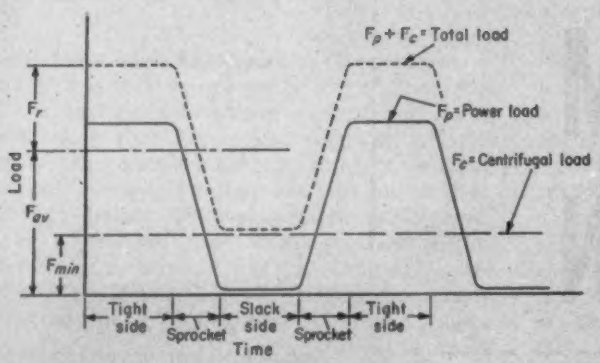


Fig. 3—Periodic load on a chain link due to power transmitted (solid line) and centrifugal action

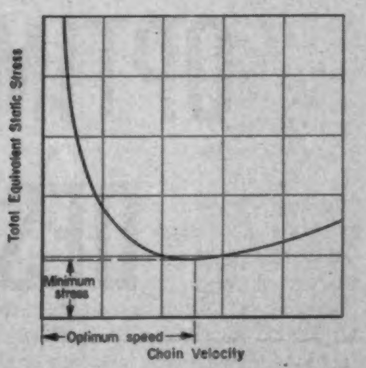


Fig. 4 — General relationship between stress and velocity for a chain transmitting constant horsepower

steps 2 and 3 depending on each other so that successive trials often are required.

Introduction of a fourth step

4. Determine the optimum speed

does not simplify design. Steps 2, 3 and 4 are now dependent on each other and more successive trials may be required. Sometimes the speed is specified and cannot be altered, in which event step 4 cannot be considered, but in a large number of cases the speed can be varied over a large range. The present article deals with this class of machines.

The term *optimum speed* can have many meanings. It can mean the speed for greatest economy, the speed that produces the smallest deflection of a critical element, the speed that produces the lowest overall stress pattern, the speed for minimum vibration, or any other meaning that is important to the job at hand. This article deals largely with stress effects, leading to increased power or to lighter weight.

One method of determining the optimum speed is to assume a series of speeds and design a machine for each of these speeds. The one that is optimum according to the chosen definition is selected. This process has been going on haphazardly and often the designers were probably not fully conscious of what they were doing. The method of designing machines for a series of assumed speeds may be simplest in some cases. In others an analytical investigation of the effect of varying the speed can be advantageous. The class of machines that do not have to be designed for a specified speed is so large that no general rules can be laid down.

Fatigue Loading: A typical type of loading is shown in *Fig.* 1, where the load varies periodically from F_{max} to F_{min}. This is also shown as an equivalent static load F_{av} and a completely reversed load F_r superimposed upon it. Resistance of materials to static loads is measured by yield strength, and to completely reversed periodic loads by the endurance limit. For combinations of static and reversed loading, experience shows that the linear relationship proposed by Soderberg and shown in *Fig.* 2 is conservative.[1] Plotting actual stress due to static load as abscissa and actual stress due to reversed load as ordinate establishes a point such as *F*, *Fig.* 2. With *F* below Soderberg line, design is safe.

For safe working stress relationships a parallel to the Soderberg line may be drawn as shown in *Fig.* 2, using the factor of safety, *N*. If loads are tensile, actual static stress is F_{avr}/A for ductile materials and reversed stress is kF_r/A, where *k* is the stress-concentration factor. From similar triangles

$$\frac{\dfrac{s_y}{N} - \dfrac{F_{av}}{A}}{s_y} = \frac{k\dfrac{F_r}{A}}{s_e}$$

which may be written

$$\frac{s_y}{N} = \frac{F_{av} + \dfrac{s_y}{s_e} k F_r}{A} \qquad (1)$$

This relationship simply converts a reversed load to an equivalent static load.

Bending loads with static and reversed components may be similarly analyzed, with this result:

$$\frac{s_y}{N} = \left(M_{av} + \frac{s_y}{s_e} k M_r \right) \frac{C}{I} \qquad (2)$$

and for torsion loads

$$\frac{(s_s)_y}{N} = \left(T_{av} + \frac{s_y}{s_e} k_t T_r \right) \frac{C}{J} \qquad (3)$$

in which the ratio s_y/s_e can be used because it is nearly equal to $(s_s)_y/(s_s)_e$.

In the examples which follow the methods are general, and any stress-load relationship could be used. The Soderberg line of *Fig.* 2 is considered safe though some designers prefer the well-known but less conservative Goodman line, *Fig.* 2. A more

1. Joseph Marin—*Engineering Materials*, Prentice-Hall Inc., New York, 1952, Page 187.

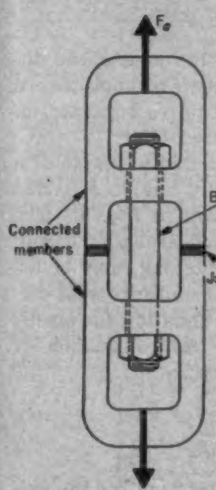


Fig. 5—Typical bolted assembly. With metal-to-metal joint the connected members usually are stiffer than the bolt; a gasket at the joint makes connected assembly less stiff

complicated relationship requires more work to solve a problem, but treating the reversed load as an equivalent static load makes it possible to plot it on the same axis with the static load, leaving the other axis free to be used with some other variable. This is particularly valuable in optimum design as it greatly simplifies the work.

Optimum Chain Speed: A power chain is subjected to two principal forces—power transmission and centrifugal. Evidently, as the speed approaches zero the centrifugal force approaches zero and, for constant horsepower, the power force approaches infinity. As the speed approaches infinity so does the centrifugal force, but the power force approaches zero. For a particular chain and power there is an intermediate speed at which the total stress due to static and reversed loading is a minimum. This optimum velocity can be obtained through the use of Equation 1, as follows.

Centrifugal tension on a chain is constant for a given speed and equal to

$$F_c = \frac{qv^2}{32.2} \qquad (4)$$

where q is the weight of the chain, lb per ft; v is chain velocity, ft per sec; and 32.2 is gravity acceleration, ft per sec per sec.

Power force on the link is maximum on the tight side and zero on the slack side. On the tight side

$$F_p = \frac{550P}{v} \qquad (5)$$

where P is the power, horsepower units.

These forces are illustrated in *Fig.* 3, the power force being shown as changing gradually while the link passes around the sprocket. Although variable load curves are usually shown as harmonic, *Fig.* 1, it has been found that the shape of the curve has little effect on the endurance.

From *Fig.* 3 it is evident that

$$F_{av} = F_c + \frac{F_p}{2}$$

$$F_r = \frac{F_p}{2}$$

When values from Equations 4 and 5 are substituted in the foregoing and then in Equation 1, the total equivalent static stress becomes

$$s = \frac{1}{A}\left[\frac{qv^2}{32.2} + \frac{550P}{2v}\left(1 + \frac{s_y}{s_e}k\right)\right] \qquad (6)$$

A curve of s plotted against v has the general shape indicated in *Fig.* 4. Differentiating s with respect to v and equating to zero,

$$\frac{ds}{dv} = \frac{1}{A}\left[\frac{2qv}{32.2} - \frac{550P}{2v^2}\left(1 + \frac{s_y}{s_e}k\right)\right] = 0$$

It is evident that the second derivative would be positive; hence the value of v at which the first derivative is zero gives the minimum value of s. Solving for this optimum chain velocity:

$$v_{opt} = 16.4\sqrt[3]{\frac{P}{q}\left(1 + \frac{s_y}{s_e}k\right)} \qquad (7)$$

Equation 7 was derived by Stamets[2] in a somewhat different manner. He suggests that $1 + (s_y/s_e)k$ can be taken as 5 for steel silent chain. The effect of dynamic loading is not included in Equation 7, but could be by modifying the factor $1 + (s_y/s_e)k$. Optimum speeds given by this analysis are high and require fairly large sprockets, thus reducing chordal action and impact effects.[2]

Optimum Bolt Load: An important example of optimum design technique is the determination of

2. W. K. Stamets Jr.—"Dynamic Loading of Chain Drives," *ASME Transactions*, Vol. 73, 1951, Page 655.

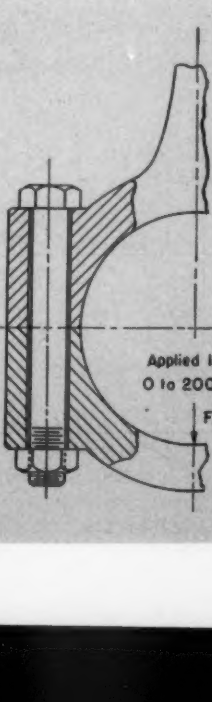


Fig. 6 — Bolted connecting-rod end subject to alternating load

proper initial tightening load for bolted assemblies subjected to repeated loading. Optimum initial tightening or setting-up load is that which gives minimum equivalent static load (Equation 1) for a given external variable load.

A bolted assembly is pictured in *Fig.* 5. Depending on the details of the application the assembly may or may not include a gasket at the joint faces. Presence of a gasket greatly increases the relative compliance (deformation per unit load) of the connected members in relation to that of the bolt itself, but does not affect the principles involved.

When the bolt is first tightened, and before the external load is applied, the tension in the bolt and the compression in the connected members are equal. Corresponding deformation of the bolt is $F_i e$ and of the connected members is $F_i c$, where F_i is the initial load, e is the elongation of the bolt per unit load and c is the compression of the connected members per unit load.

Application of the external applied load F_a results in a final bolt load F which is generally less than $F_i + F_a$, and a final compressive load F_c in the connected members. The change in length of bolt and connected members is equal to the final bolt length minus the initial, or $Fe - F_i e$. Final deformation of the connected members is then $F_i c - (F - F_i)e = F_c c$. Compressive load in the connected members therefore is

$$F_c = F_i - \frac{e}{c}(F - F_i)$$

Final tension in the bolt is the sum of the applied external load and the final load in the connected members:

$$F = F_c + F_a = F_i - \frac{e}{c}(F - F_i) + F_a$$

from which

$$F = \frac{c}{c + e} F_a + F_i \quad \text{.........................} \quad (8)$$

Equation 8 does not hold when the joint opens, inasmuch as the connected members cannot take a tensile load. This condition requires separate investigation and will be considered later.

Connecting-Rod Bolt: A practical example of a bolted assembly, which is readily analyzed, is the connecting rod or link end shown in *Fig.* 6. The cylindrical portion of the rod end around the bolt, which is compressed by the tightening, may be assumed to be a hollow cylinder of uniform cross-sectional area. From the familiar equation for elastic deformation, $\delta = Fl/AE$, the unit deflection or stretch per unit load for the bolt is

$$e = \frac{\delta}{Fl} = \frac{1}{A_b E_b}$$

and the unit compression per unit load for the hollow cylindrical portion of the connecting rod is

$$c = \frac{\delta}{Fl} = \frac{1}{A_c E_c}$$

from which the factor $c/(c + e)$ in Equation 8 may be written

$$\frac{c}{c + e} = \frac{A_b E_b}{A_b E_b + A_c E_c} \quad \text{.................} \quad (9)$$

Solving for the initial tightening load from Equations 8 and 9

$$F_i = F - \frac{A_b E_b}{A_b E_b + A_c E_c} F_a \quad \text{..........} \quad (10)$$

Although Equation 10 holds only so long as the joint remains closed, it includes the condition when the joint is on the verge of opening, in which event the final load is equal to the external applied load,

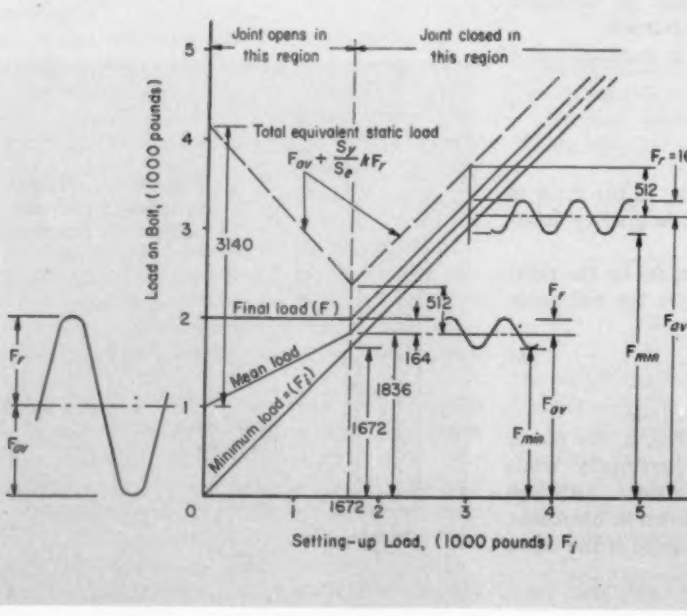


Fig. 7—Bolt load versus setting-up load for the assembly shown in *Fig.* 6 when subjected to an alternating external load from 0 to 2000 lb. Optimum setting-up load of 1672 lb results in minimum total equivalent static load on bolt equal to 2348 lb. Actual setting-up load should be somewhat higher to provide safe margin against joint opening

$F = F_a$. The initial load that will permit this condition is, from Equation 10,

$$F_i' = \left[1 - \frac{A_bE_b}{A_bE_b + A_cE_c}\right]F_a = \frac{A_cE_c}{A_bE_b + A_cE_c}\,F_a \quad (11)$$

EXAMPLE 1: The connecting rod end in *Fig.* 6 is subjected to an externally applied load which varies from 0 to 2000 lb. Each bolt is a ½—20 NF with shank area 0.196 sq in., root area 0.149 sq in., stress-concentration factor $k = 2.5$. The steel of which it is made has $s_y = 35{,}000$ psi and $s_e = 28{,}000$ psi. The cylindrical portion of the connecting rod surrounding each bolt has area 1 sq in. and the material also is steel. To what optimum load should the bolt be initially tightened?

From Equation 11, the initial tightening that will cause the joint to be on the verge of opening is

$$F_i' = \frac{1}{1 + 0.196}\,(2000) = 1672 \text{ lb}$$

This value is plotted in *Fig.* 7. As the load on the connecting rod varies from 0 to 2000 lb the load on the bolt varies from 1672 to 2000 lb, while F_{av} is 1836 lb and F_r is 164 lb. The reversed component in terms of equivalent static load (Equation 1) is:

$$k\,\frac{s_y}{s_e}\,F_r = (2.5)\,\frac{35{,}000}{28{,}000}\,(164) = 512 \text{ lb}$$

For this bolt a reversed loading of 164 lb is as destructive as a static load of 512 lb.

For initial tightening loads higher than 1672 lb, the final load may be found from Equation 10, which becomes

$$F = F_i + \frac{0.196}{0.196 + 1}\,(2000) = F_i + 328$$

The second term (328 lb) represents twice the reversed load in the bolt. In a given application this term is constant so long as the joint remains closed; hence the final load curve is parallel to the minimum load line in this region. Over-tightening a bolt does not change the variable load component.

In the region where the joint opens the final load will always be F_a, in this example, 2000 lb. If the nut is just brought up snug, that is, the initial setting-up load is zero, $F_{av} = F_r = \frac{1}{2}F_a = 1000$ lb. The equivalent static load for F_r is

$$(2.5)\,\frac{35{,}000}{28{,}000}\,(1000) = 3140 \text{ lb}$$

These values are plotted in *Fig.* 7. The mean load always lies midway between the final load and the minimum load. Since the final and minimum load curves are straight lines the mean load and the total equivalent static load curves will also be straight lines.

It can be seen from *Fig.* 7 that the optimum setting-up load is that load which causes the joint to be on the verge of opening when the maximum external load is applied. It can also be seen from *Fig.* 7 that the total equivalent static load curve for setting-up loads ranging from zero to the condition where Hooke's law fails to hold can be obtained by determining the values corresponding to zero and the optimum.

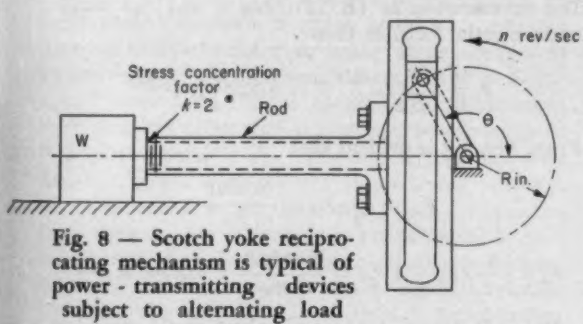


Fig. 8 — Scotch yoke reciprocating mechanism is typical of power - transmitting devices subject to alternating load

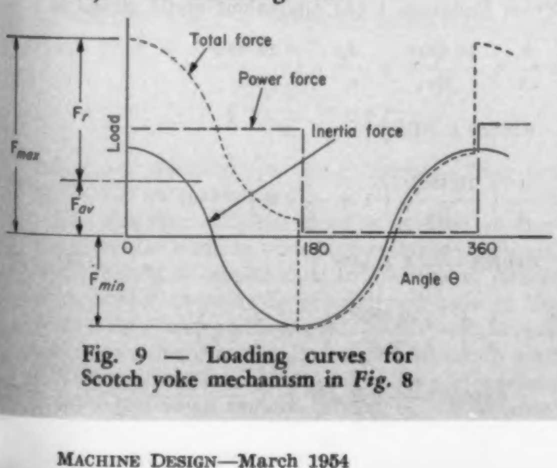


Fig. 9 — Loading curves for Scotch yoke mechanism in *Fig.* 8

Fig. 10—Total equivalent static stresses for two mechanisms of different size both transmitting the same power

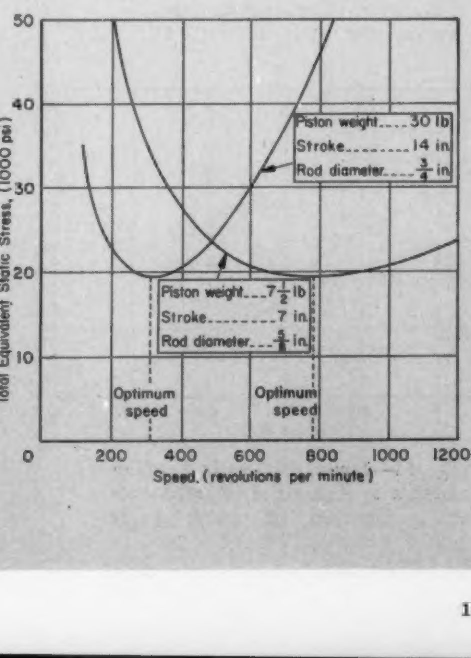
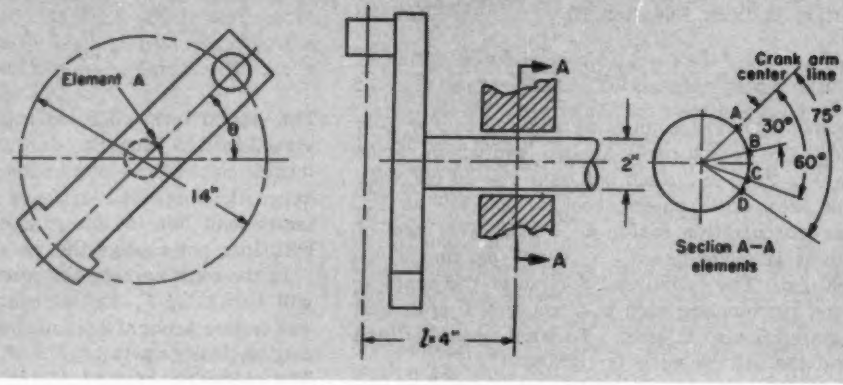


Fig. 11—Crank and crankshaft element of Scotch - yoke mechanism in *Fig.* 8



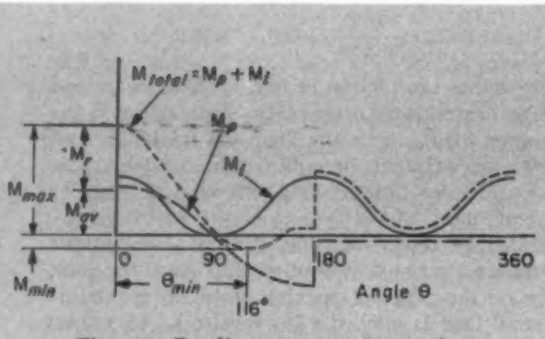


Fig. 12—Bending moment curves for element *A* of crankshaft in *Fig.* 11

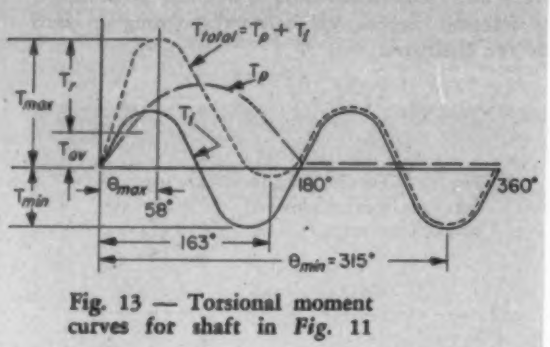


Fig. 13 — Torsional moment curves for shaft in *Fig.* 11

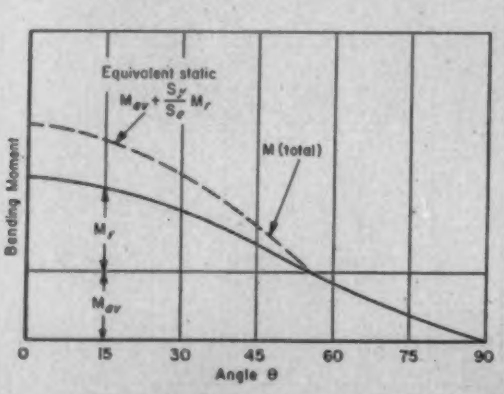


Fig. 14—Equivalent static bending moment at element *A* of crankshaft as a function of crank angle

Optimum Speed of Mechanisms: A large class of mechanisms involves linkages subjected to power transmission forces plus periodic inertia forces due to reciprocating or oscillating motion. A typical example is the Scotch yoke shown in *Fig.* 8. Here it is assumed that the power force is constant over the first half of each cycle and zero over the remaining half. If an adequate flywheel is used the speed will be very nearly constant throughout a cycle. The loading on the horizontal rod will be investigated. The shapes of the power force and inertia force curves for this member are indicated in *Fig.* 9. The speed is $n = \omega/2\pi$ revolutions per second and during each revolution the force acts through $2R/12$ feet. The power force through each half cycle is then

$$F_p = \frac{550\,P\,(2\pi)}{\omega}\,\frac{12}{2R} = 20{,}800\,\frac{P}{R\omega}$$

The acceleration is $(R/12)\omega^2\cos\theta$ and the maximum inertia force is then

$$F_i = \frac{W}{32.2}\,\frac{R}{12}\,\omega^2 = 0.00258\,R\,\omega^2 W$$

From *Fig.* 9 it is evident that

$$F_{av} = \frac{1}{2}\,F_p = \frac{10{,}400P}{R\omega}$$

$$F_r = \frac{1}{2}\,(F_p + 2\,F_i) = \frac{10{,}400P}{R\omega} + 0.00258\,R\,\omega^2 W$$

From Equation 1 the equivalent static stress is

$$s = \frac{1}{A}\left\{\frac{10{,}400P}{R\omega} + \frac{s_y}{s_e}\,k\left(\frac{10{,}400P}{R\omega} + 0.00258\,R\,\omega^2 W\right)\right\}$$

$$= \frac{1}{A}\left\{\frac{10{,}400P}{R\omega}\left(1 + \frac{s_y}{s_e}\,k\right) + 0.00258\,\frac{s_y}{s_e}\,k\,R\,\omega^2 W\right\} \qquad (12)$$

$$\frac{ds}{d\omega} = \frac{1}{A}\left\{-\frac{10{,}400P}{R\omega^2}\left(1 + \frac{s_y}{s_e}\,k\right) + 0.00516\,\frac{s_y}{s_e}\,k\,R\,\omega\,W\right\}$$

Inasmuch as the second derivative, $d^2s/d\omega^2$, would be positive, setting $ds/d\omega$ equal to zero gives the value of ω for minimum s. When this is done,

$$\omega = 126\sqrt[3]{\frac{P}{R^2W}\left(1+\frac{s_e}{s_y k}\right)} \quad \ldots\ldots\ldots\ldots\ldots (13)$$

EXAMPLE 2: For a mechanism such as in *Fig.* 8 with $W = 30$ lb, $R = 7$ inches, $P = 20$ hp, material steel having $s_y = 50{,}000$ psi and $s_e = 30{,}000$ psi, and stress-concentration factor at thread $k = 2$, the value of ω from Equation 13 is 32.8 rad per sec or 314 rpm. If the rod is threaded ¾—10 NC the cross-sectional area at the thread roots is 0.3 sq in. and the stress at optimum speed is, from Equation 12, 19,500 psi. How stress varies with speed is shown in *Fig.* 10.

For an alternative design with the crank radius R reduced to 3.5 inches and the weight to 7.5 lb, the optimum speed, from Equation 13, becomes 785 rpm. For the same stress, 19,500 psi, the rod area could be reduced to 0.238 sq in., or a ⅝—18 NF thread. These changes thus result in a smaller lighter machine with higher optimum speed for the same power capacity, *Fig.* 10. Whether such changes are feasible depend on the nature of the problem.

SPEED AND SIZE RELATIONSHIPS: The foregoing example, and *Fig.* 10, demonstrate that optimum speed increases as the size of a machine or mechanism decreases. With optimum speed established for a particular configuration and stress condition, it is possible with the aid of Equation 13 to set up relationship between speed, size, power capacity, etc., for an entire line of geometrically similar machines.

Denote the size of the machine or mechanism by some basic dimension d which could be a cylinder bore, a shaft diameter or some other convenient measurement. Then all weights such as W in Equation 13 are proportional to d^3, linear dimensions such as R are proportional to d, and power capacity P is proportional to $d^3\omega$. The last relation is easily derived: power equals force times linear velocity, but force equals stress times area; therefore for the same stresses, power forces are proportional to area and therefore to d^2; also, linear velocity is proportional to linear dimension times angular velocity and therefore is proportional to $d\omega$. Substituting in Equation 13 the foregoing relationships, i.e., $W \propto d^3$, $R \propto d$ and $P \propto d^3\omega$,

$$\omega \propto \sqrt[3]{\frac{d^3\omega}{d^2d^3}} \text{ or } \omega \propto \frac{1}{d}$$

All stress relationships being identical, they can be treated as constants.

The fact that optimum speed is inversely proportional to size leads to some interesting conclusions. While weight is proportional to the cube of dimensions, power capacity is proportional only to the square of linear dimensions; hence weight per horsepower increases in linear proportion to size, a fact to be remembered in comparing performance of large and small engines, pumps, or compressors,

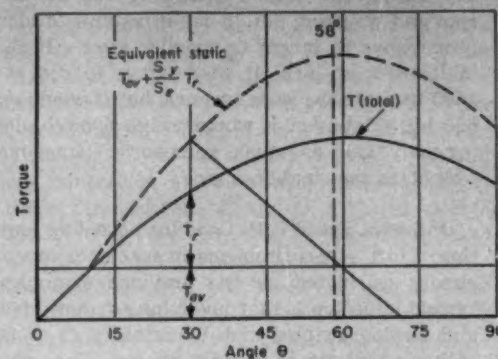
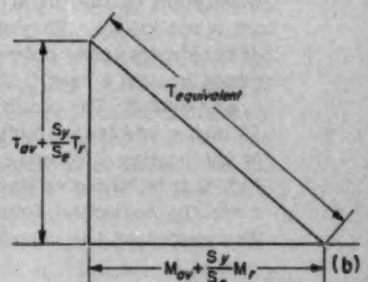


Fig. 15 — Above — Equivalent static torsional moment for crankshaft as a function of crank angle

Fig. 16—*(a)* Equivalent static combined torsional and bending moment derived from *Figs.* 14 and 15 as shown at *(b)*. Diagram *(b)* can be conveniently constructed on *Fig.* 15



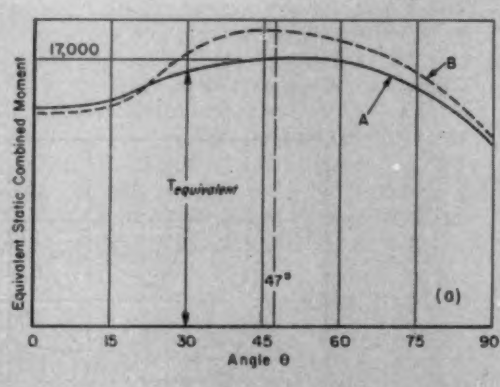
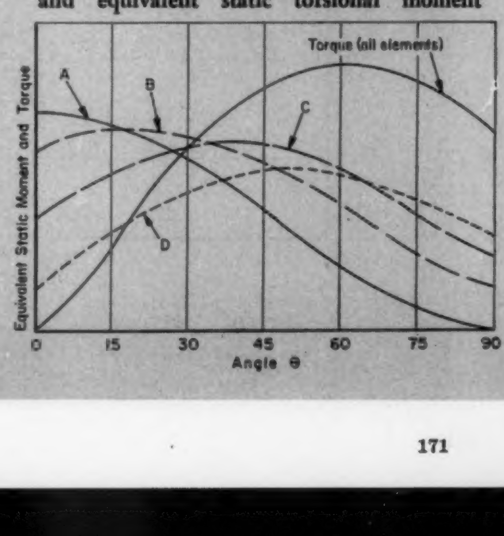


Fig. 17 — Below — Equivalent static bending moments for elements *A*, *B*, *C* and *D*, and equivalent static torsional moment



to mention only a few typical machines.

As an example, it is conceivable that a present-day automobile engine delivering 150 hp at 3600 rpm and weighing 750 lb could be built to identical proportions in larger or smaller sizes. If it were built twice as large it would have to run at only 1800 rpm for the same stresses but it would deliver 600 hp. However, it would weigh 6000 lb, delivering only half as much horsepower per pound of weight as the smaller engine.

Optimum Speed with Combined Bending and Torsion: Most stress problems in machines are not so simply calculated as the previous examples. A typical situation is that involving combined bending and torsion coupled with uncertainty as to the location where stress is most severe. The overhung crankshaft shown in *Fig.* 11 is such an example. Consideration shows that maximum stress probably occurs at the center of the bearing, but at what point on the surface of the shaft (*A, B, C,* or *D*) is not known. The following numerical solution demonstrates how optimum speed can be determined in such a case.

EXAMPLE 3: The crank in *Fig.* 11 has a stroke of 14 inches and is overhung 4 inches from the center of the bearing. Forming part of the mechanism in *Fig.* 8, it is loaded as shown in *Fig.* 9, which represents the horizontal force acting on the crankpin. The shapes of the corresponding bending moment and torsional moment diagrams are shown in *Figs.* 12 and 13.

With no stress concentration, the stress due to torsion at any instant is uniform over the surface of the shaft. But for bending, element *A* on the centerline of the crank at the center of the bearing will, when $\theta = 0$, be stressed in bending higher than any other element, *Figs.* 11 and 12. This point will be investigated first. For an element at *A* the power bending moment is

$$M_p = \frac{550P(2\pi)}{\omega}\,\frac{12}{2R}\,l\cos\theta = \frac{237{,}200}{\omega}\cos\theta \text{ lb-in.} \quad (14)$$

and the inertia bending moment is

$$M_i = \frac{W}{32.2}\,\frac{R}{12}\,\omega^2\,(\cos\theta)\,(l\cos\theta) =$$

$$2.172\,\omega^2\cos^2\theta \text{ lb-in.} \quad (15)$$

For any element on the shaft surface at the center of the bearing the power torsional moment is

$$T_p = \frac{550P(2\pi)}{\omega}\,\frac{12}{2R}\,R\sin\theta = \frac{415{,}100}{\omega}\sin\theta \quad (16)$$

and the inertia torsional moment is

$$T_i = \frac{W}{32.2}\,\frac{R}{12}\,\omega^2\,(\cos\theta)\,(R\sin\theta) =$$

$$3.8\,\omega^2\sin\theta\cos\theta \quad (17)$$

Because any attempt to determine optimum speed

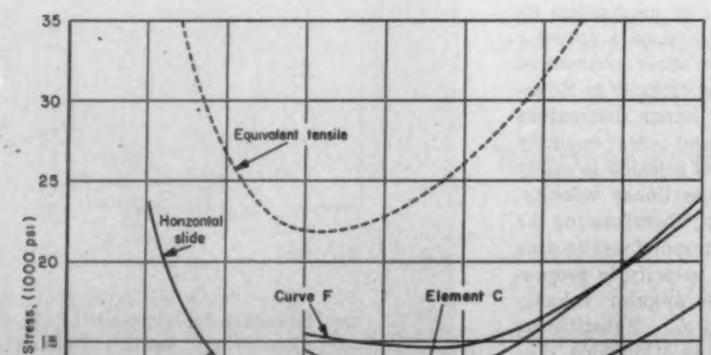


Fig. 18 — Equivalent static combined shear stresses for each element, as a function of speed

analytically becomes highly involved, it must be solved by a series of trials. For a first assumption $\omega = 50$ rad per sec. *Fig.* 12 shows the bending moment for element *A* at this speed, and *Fig.* 13 the torsional moment curves. In *Fig.* 12 the important data are:

$$M_{max} = 10{,}180 \text{ lb-in. at } \theta = 0$$
$$M_{min} = -1040 \text{ lb-in. at } \theta = 116 \text{ deg}$$
$$M_{av} = 4570 \text{ lb-in.}$$
$$M_r = 5610 \text{ lb-in.}$$

In *Fig.* 13 the important data are:

$$T_{max} = 11{,}320 \text{ lb-in. at } \theta = 58 \text{ deg}$$
$$T_{min} = -4750 \text{ lb-in. at } \theta = 163 \text{ deg}$$
$$T_{av} = 3285 \text{ lb-in.}$$
$$T_r = 8035 \text{ lb-in.}$$

For combined bending and torsion the equivalent maximum torsion is given by the well known relationship $T_e = \sqrt{M^2 + T^2}$. In the absence of stress concentration, the corresponding equivalent static shear stress (see Equations 2 and 3) is

$$s_s = \frac{C}{J}\sqrt{\left(M_{av} + \frac{s_y}{s_e}M_r\right)^2 + \left(T_{av} + \frac{s_y}{s_e}T_r\right)^2} \quad (18)$$

The actual total moment curve and the average moment from *Fig.* 12 are shown replotted on *Fig.* 14 together with the equivalent static moment. The corresponding torque curves are shown in *Fig.* 15. The combined curve corresponding to the term under the radical in Equation 18 is plotted in *Fig.* 16*a*. It may be constructed graphically as shown in *Fig.* 16*b*. Maximum equivalent static torque, 17,000 lb-in., occurs when $\theta = 47$ deg. The corresponding shear stress in the 2-inch diameter shaft is, from Equation 18, 10,800 psi.

For element *B*, *Fig.* 11, the moments are calculated from Equations 14 and 15 but allowing for the 30-degree displacement of element *B* from element *A*, giving

$$M_p = \frac{237{,}200}{\omega}\cos(\theta - 30°)$$
$$M_i = 2.172\,\omega^2 \cos\theta \cos(\theta - 30°)$$

OPTIMUM DESIGN

The equivalent static bending moment for this element is plotted in *Fig.* 17. It is evident that the maximum is less than for element *A* but the bending moment peak is shifted toward the torque peak, resulting in a higher equivalent static combined moment, *Fig.* 16*a*, and a higher combined stress. Equivalent static bending moment curves for the other elements, *C* and *D*, are also plotted in *Fig.* 17. The corresponding stresses are plotted in *Fig.* 18.

Repeating the entire procedure for several other speeds gives enough points to draw curves for maximum combined stress for each point as a function of speed, *Fig.* 18. The optimum speed appears to be between 40 and 50 rad per sec.

The stress-speed curve for the rod in the same mechanism, *Fig.* 8, is plotted from *Fig.* 10. However, this is a tensile stress whereas the stresses for elements *A*, *B*, *C* and *D* are shear. On the assumption that the shear yield strength is half the tensile yield, the rod stress has been replotted at half the actual value for comparison. The optimum speed when both members (rod and shaft) are considered is at the intersection of the curves for the rod and for the crank element of maximum stress, at $\omega = 48$ rad per sec, approximately.

For simplicity it is often assumed that the maximum bending moment and maximum torque occur simultaneously. In the present example this would eliminate much of the previous work, inasmuch as it would be unnecessary to consider several elements, *A*, *B*, *C* and *D*, or to plot the curves in *Fig.* 17. The maximum total torque and maximum total bending moment could be found analytically from Equations 14, 15, 16 and 17 by differentiating the expressions for $(M_p + M_i)$ and $(T_p + T_i)$ with respect to θ and setting equal to zero, solving for θ and so determining the maximum values. Follow-

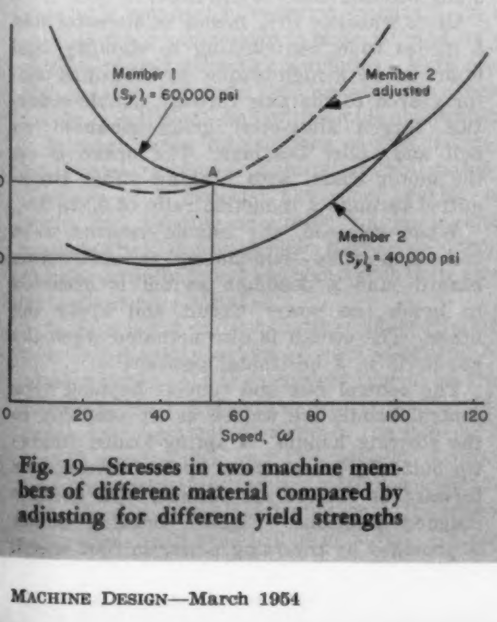


Fig. 19—Stresses in two machine members of different material compared by adjusting for different yield strengths

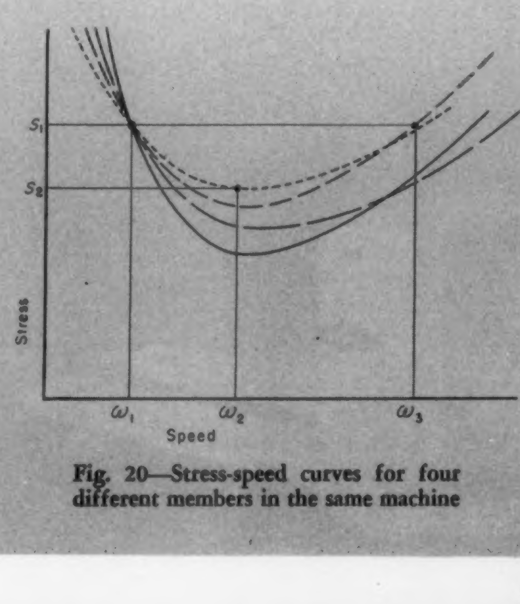


Fig. 20—Stress-speed curves for four different members in the same machine

ing this procedure results in curve *F*, *Fig.* 18, which is well on the side of safety.

If an optimum speed for several members is to be obtained in this manner and they are made of materials having different yield points, then it is necessary to adjust them to one yield point. An example is shown in *Fig.* 19. Stress-speed curves are plotted for two members, 1 and 2, having yield points of 60,000 and 40,000 psi, respectively. The curve for member 2 is increased by the factor 60,000/40,000. The optimum speed is given by the intersection *A*. For this speed the factor of safety for both members is

$$\frac{60{,}000}{38{,}000} = \frac{40{,}000}{25{,}700} = 1.56$$

Any number of curves can be adjusted in this manner to a chosen stress.

The ideal machine is often considered to be one in which the factors of safety for all members are equal. The stress-speed curves for the members of such a machine appear in *Fig.* 20. Here the machine was designed to operate at speed ω_1. Any speed between ω_1 and ω_3 would reduce the stresses and the optimum would be at ω_2.

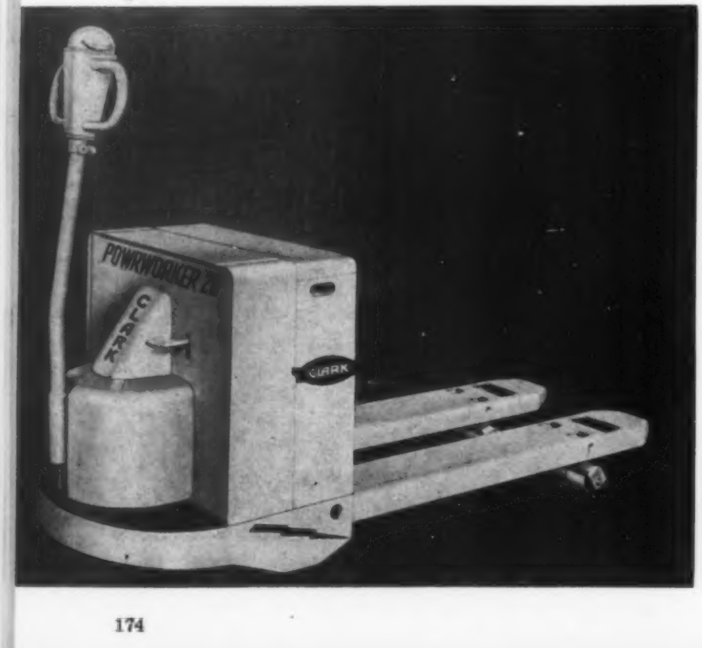
Conclusions: Problems that cannot be solved analytically can be done graphically. Sometimes problems that can be done analytically can be done more easily graphically. In examples 2 and 3 the power load was assumed to change instantaneously from zero to a maximum. This is a suddenly applied load and theoretically produces a stress that is double that of a gradually applied load. This condition, and shock, can be included on the curves.

Whether it is better to over or under-tighten a screw fastening subjected to variable loading can be determined from the slopes of the two branches of the total equivalent static load curve, *Fig.* 7. For practical reasons it is usually better to over-tighten. This may be necessary to prevent fluid leakage and the screw fastening is less likely to work loose. If it is over-tightened a certain amount of loosening will lower the stress. If it is under-tightened loosening will increase the stress. The method described in this article should not be considered as a universal method for all designs but it can be considered as an additional tool that in some cases can be used for making better designs.

CONTEMPORARY DESIGN

Powered Hand Truck

SHORT overall length and maximum load stability are features of the new Powerworker 26 line of powered hand trucks developed by Clark Equipment Co. Total length of the truck is only 26 inches larger than the load forks. Low center of gravity, resulting in a high degree of load stability, is gained by locating the battery in an underslung position close to the floor.



Drive wheel is $10\frac{1}{2}$ inches in diameter and 6 inches wide, contributing to stability, and is driven by a high-torque, series-wound motor geared to the axle through double-reduction forged alloy-steel gears mounted on ball and roller bearings. The brake is on the motor shaft, with braking effort transmitted through a reduction ratio of 22 to 1.

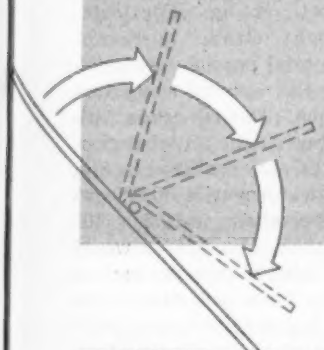
When released, the handle returns to a vertical position—eliminating an obstruction hazard—and a deadman switch is actuated to break the power circuit and apply the brake. The switch is also actuated when the handle is in a horizontal position.

The control box and tubular bayonet type control handle are welded as an assembly to the steering handle. A spring-loaded fingertip butterfly lever selects speeds; two speeds forward or reverse are possible with definite notched positions. Electric braking action is provided by reversing power in first speed.

ductility and plastic deformation

Utilizing ductility properties in comparing and evaluating engineering materials

By Robert L. Stedfeld *Associate Editor, Machine Design*



PRECISE meaning of ductility, and effective use of common ductility properties for evaluating a material, have long been a problem in design. In years past, values for elongation and reduction in area, the two most common ductility properties, have been regarded as indicators of the effectiveness of a material in resisting overloads, as measures of the toughness of a material, and as rough criteria of cold working capabilities. More recently, the reasons for these assumptions have been severely questioned—and with ample justification.

In most cases, assumptions of this nature have been made because of intuitive realization that ductility of a material has some bearing on capacity for resisting overloads, toughness, and cold workability. But a higher degree of correlation than is actually found has sometimes been assumed, with disastrous results.

In comparing and evaluating different materials, how can ductility values be used effectively? The answer varies with the expected type of service, but relatively clear distinctions can be made. No matrial can be evaluated except in the light of its intended service use, and this precept is especially true for ductility properties. Mere specification of high ductility does not insure that the characteristics actually desired will be achieved. This article will attempt to clarify the design areas in which ductility values can or cannot be accepted as indicating the serviceability of the material, will summarize typical ranges for engineering materials, and will point out the variations which can occur in the materials themselves.

What Is Ductility?: Ductility is a word that everybody uses and knows about, but nobody understands—largely because it is not an exact engineering description of any basic characteristic of a material. Dictionary definitions are vague, and the word is subject to as many interpretations as there are people to interpret it.

Ductility can be considered as a measure of the plastic deformation characteristics of a material, or the ability of the material to be deformed without rupture by relatively slow application of load.

Conventionally, ductility of a material is measured by two properties—elongation and reduction of area—both obtained in the course of a standard tensile test. Definitions are:

Elongation: Increase in length between gage marks on a standard tensile test specimen after fracture, expressed as a percentage of the original length.

Reduction of Area: Difference between area of

Photo, courtesy American Brass Co.

High ductility is needed for severe forming operations in making bellows. Material is brass, cupped from a round blank and drawn to a long tube with walls about 0.005-inch thick, after which bellows are formed by roll-forming or by hydraulic expansion outward into a collapsible die. Cumulative effect of this cold working hardens and strengthens the material

smallest cross section of a fractured specimen and the original area, expressed as a percentage of the original area.

Other tests are also used to measure ductility of engineering materials, but the two values resulting from the standard tensile test are by far the most prevalent. Thus, angle of twist in a torsion test and deflection at fracture in flexure tests for brittle materials are both ductility indicators. Cold-bend tests such as the Olsen, Tour-Marshall or Scheler, and cupping tests like the Erichsen or Olsen essentially measure ductility although these tests have been designed to attempt to gage the cold workability of materials.

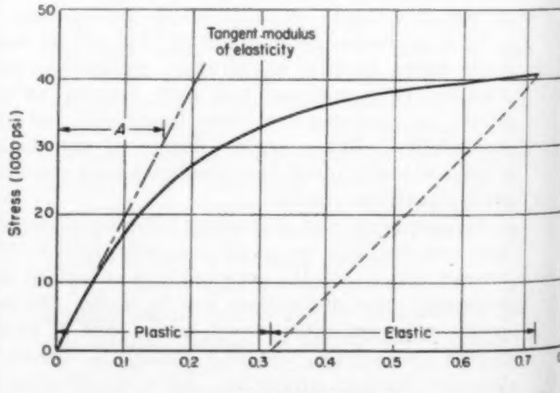
Elastic and Plastic Elongation: Elongation values from a tensile test are measures of the *plastic* deformation of a test piece, as distinguished from the *elastic* deformation of a material up to its yield point. Elongation figures are usually obtained by fitting together the ends of the fractured specimen and measuring the increase in gage length—after the material has recovered the elastic part of its deflection in the test.

The point is illustrated in *Fig.* 1, which shows the stress-strain curve for a soft gray iron.[1] If the tensile test is carried to a point just short of fracture (approximately 41,000 psi), the specimen would elongate 0.722 per cent. If the load is then released, the specimen recovers 0.404 per cent (elastic strain) and is left with a permanent deformation of 0.318 per cent (plastic strain). This plastic deformation is the value reported as "elongation" in tensile-test figures. Obviously total capacity for deformation is much larger than the elongation value.

Magnitude of either of these two values is still small compared with the much higher elongation values normally encountered in ductile materials. For most materials, elongation values obtained by autographic stress-strain recorders (essentially total strain) are usually considered to be synonymous with those obtained by measuring the specimen after fracture (plastic strain only). For steels, the elastic strain is considered to be negligible for total strains greater than 5 per cent. In designing for strain rather than stress, however, this assumption is often not justified, as will be shown later. Further consideration of elastic and plastic strain sometimes helps explain why materials with what seem to be hopelessly low elongation can still prove to be extremely useful in design.

One further point is worth noting. For materials with a relatively straight "elastic" portion of the stress-strain curve, the total possible deflection can be estimated by adding the reported elongation figure to the elastic deflection. For materials with a curved stress-strain diagram in the initial region, such is not the case. For the gray cast iron in *Fig.* 1, the 0.1 per cent offset yield stress is 30,600 psi and the tangent modulus of elasticity is 19.5×10^6 psi. Elastic strain using these two figures would be $30{,}600/(19.5 \times 10^6) = 0.00157$ inch per inch or 0.157 per cent (*A* in *Fig.* 1). Total deflection would be $0.157 + 0.318 = 0.475$ per cent compared with an actual 0.722 per cent total observed deflection. Use of secant modulus of elasticity, in which the modulus line is drawn from the zero point to some

Fig. 1—Stress-strain curve for a soft gray iron, showing elastic and plastic deformation



[1]References are tabulated at end of article.

arbitrary stress higher than the yield stress, might lead to a closer approximation, but results would still be doubtful unless the complete stress-strain curve were available.

Ductile and Brittle Failure: Actual mode of failure, either in service or on test parts, is often of prime importance in determining how the material has behaved in a particular design situation. Theoretical "property" figures, established under controlled conditions, do not take into account the wide range of variables which can affect parts in service. Much more useful information can be gleaned from the actual failure itself.

Unfortunately, certain types of failure in metal are frequently designated as "ductile" or "brittle" failures, although materials showing high elongation values may fail in a brittle manner, and sometimes even after considerable deformation. Thus, these two descriptions of failure should not be related to reported elongation values. Several ways of classifying types of fractures are:[2]

Behavior Described	Terms Used	
Crystallographic mode	Shear	Cleavage
Appearance of fracture	Fibrous	Granular
Strain to fracture	Ductile	Brittle

These two sets of classifications are not mutually exclusive. Ideally, ductile materials would have a predominantly shear type of fracture ("cone-and-cup" fracture in a tensile test) and a fibrous appearance at the break. But most fractures are mixed, showing characteristics of both types; frequently, fractures may be classified according to percentage granular or fibrous. And, as pointed out previously, ductile materials can fail with granular cleavage fractures.

RELATION TO TOUGHNESS AND IMPACT STRENGTH: So-called ductile fractures thus seem to involve more than the ductility which denotes strain-to-fracture characteristics of the metal. However, since most ductile materials exhibit fibrous fractures, there is obviously some basis for attempting to correlate strain-to-fracture characteristics with fibrous fractures.

The relationship is found in the ephemeral quality of toughness. According to more or less generally accepted definition (the word again is not a definitive engineering term), toughness is the energy required to fracture a specimen. It is thus the capacity of the material to resist fracture by a single peak load. One indicator is the area under the stress-strain curve, an integral function of strain (or ductility) and stress. Another similar gage is the "toughness index number," or the stress at fracture multiplied by the strain. Thus, ductility is evidently one of the major factors in determining whether a material fails "ductile" or "brittle."

These indicators, however, do not adequately reflect the material's ability to resist peak loads of short duration. Thus, the impact tests—Charpy, Izod and tensile—were devised to attempt to measure impact strength or toughness. These tests, similarly, give an energy measure of load absorption. They are usually used to separate materials into two different classes—brittle or ductile—according to the type of fracture (granular or fibrous), and also to estimate the toughness of the material.

Thus, from the short-duration load standpoint, "ductile" and "brittle" fractures actually become "tough" or "not-tough" fractures, respectively.

NOTCH SENSITIVITY: Impact tests, unfortunately, have been found to be subject to another infirmity —stress concentration at the base of the notch in the usual specimen. Materials have varying degrees of sensitivity to the presence of such stress

Fig. 2—Effect of section size and notch sharpness on type of fracture for fully annealed silicon-killed SAE 1025 steel

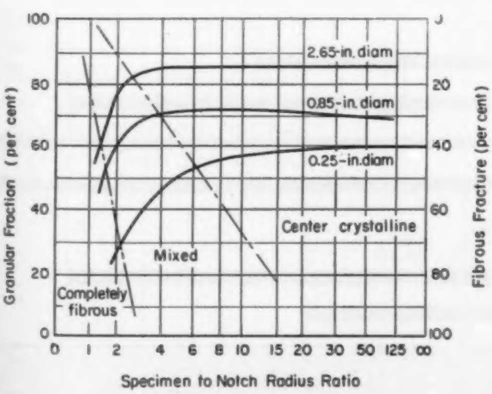
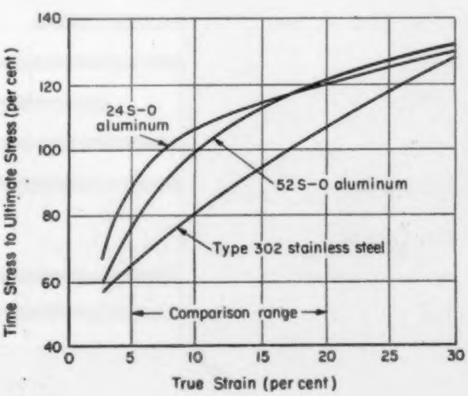


Fig. 3 — Deep-drawing criterion for comparing metals. The lower the curve in the critical strain range, the less chance of fracture at the punch radius



concentration; some normally ductile (high-elongation) materials fail in a brittle manner. Stress concentration and notch sensitivity are not confined to impact tests; presence of notches affects every type of loading as can be seen from *Fig.* 2.[3] Another factor must thus be lanced against the foregoing considerations.

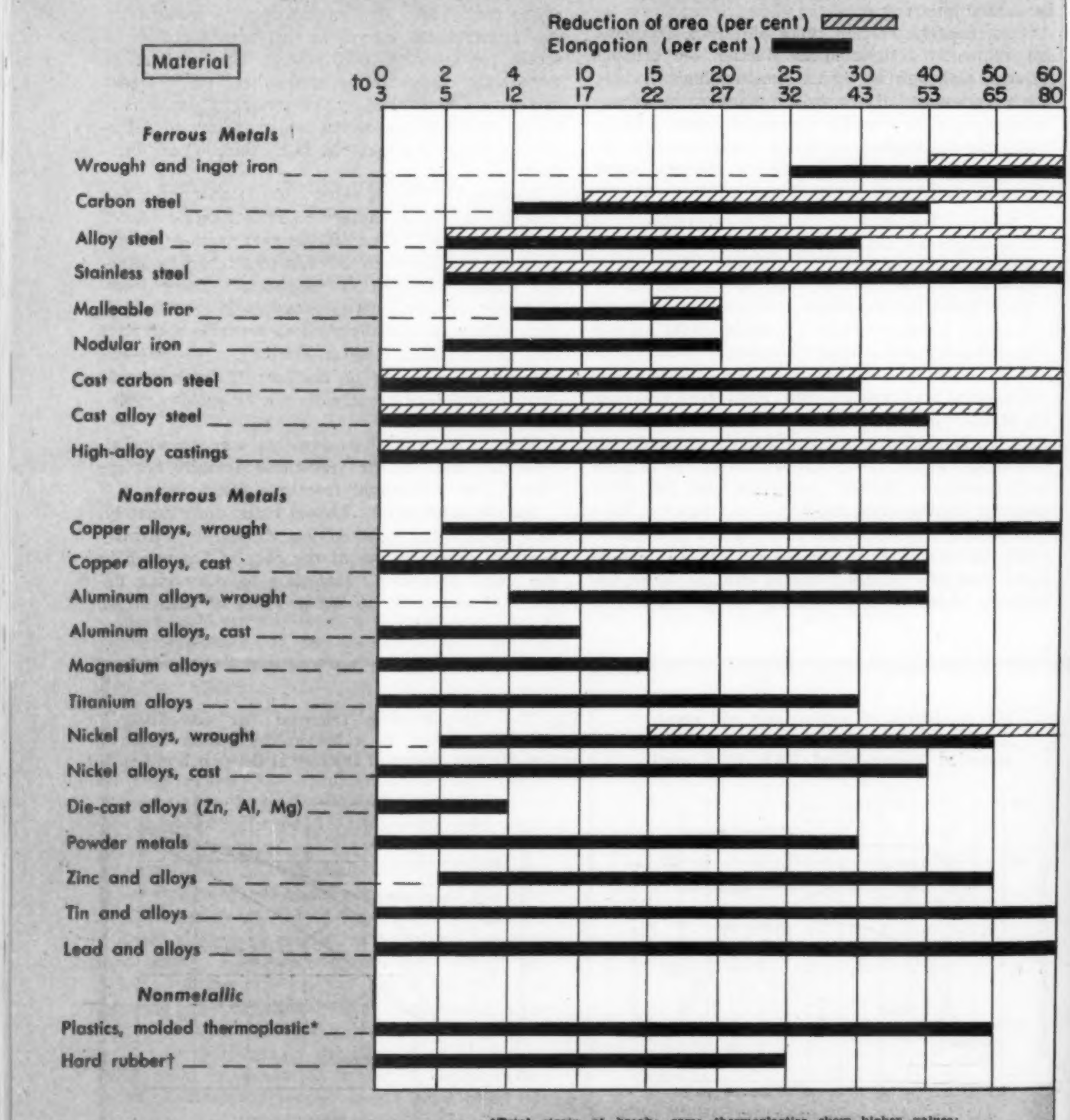
CONCLUSIONS: Whether a ductile material will fail in a "ductile" or "brittle" manner thus depends on more than just the elongation value for the material. Detailed discussion will be left for later articles; however, the main considerations determining how the material will fail, as they are known today, seem to be:

Service Conditions: Magnitude of load; direction of load; variability of load; temperature; corrosion.

Part: Size; shape; stress concentration; fits or tolerances.

Material: Strength; ductility (plastic strain to

Fig. 4—Ductility Ranges for Common Engineering Materials



*Total strain at break; some thermoplastics show higher values; typical are—polyethylene 200%, polytetrafluoroethylene 250%, nylon 300%, polyvinyl chloride 400%. †Elongation in 3 inches.

fracture); elasticity; notch sensitivity; structure (grain size, isotropy, inclusions, etc.); temperature effects; corrosion resistance.

How Ductility Values Are Used: Practical design uses of ductility values are few; elongation and reduction of area are primarily employed as rough criteria for evaluating certain physical properties. The basic engineering uses of ductility figures are:

1. Design for stress loading in the plastic range
2. Evaluating capacity for resisting overload
3. Estimating cold workability

DESIGN FOR PLASTIC DEFORMATION: There are two major ways of utilizing the ductility characteristics of a material: (1) stressing the material above its yield strength while in service to gain increased strength, and (2) compensating for misalignment by utilizing the "cold flow" characteristics of the material. The first is seldom used, since other methods of attaining higher strength by prior cold working are usually available, while the second is frequently employed.

Plastic-range design is largely an unexplored field, for obvious reasons. A certain amount of permanent deformation is involved, which must be precalculated and allowed for in the original design. This is no mean task, since it requires reasonable confidence in both the loading of the part, and the work-hardening characteristics and properties of the material. Consequently, the method is usually employed only where actual loading can be controlled by prestressing before the component is placed in service, where configuration can be controlled, or where configuration is not of prime importance.

Hoisting chain, for instance, is often prestressed before being put in service to work-harden it and raise the yield strength. To relieve internal strains, rotating parts such as turbine wheels are rotated at higher speeds than normally encountered in service, the resulting plastic flow thus redistributing and reducing residual stress. Autofrettage, subjection of a cylinder to internal hydraulic pressure to enlarge the bore about 6 per cent and the outside diameter about 1 per cent, is an effective means of increasing the elastic strength of the cylinder. In the last example, the effect is double—the elastic limit of the material is increased, and favorable residual compressive stress is induced near the bore.

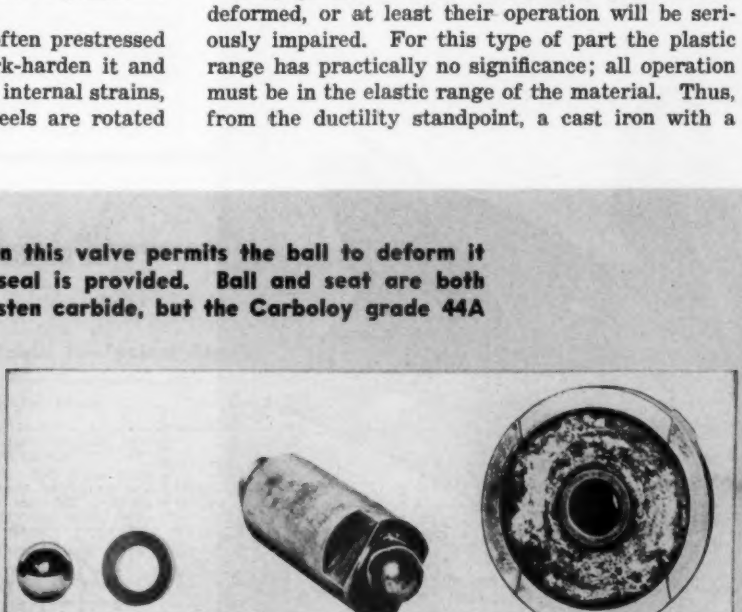
In compensating for misalignment, interference n fits, normal tolerance variations, etc., a certain amount of ductility may be needed to permit the material to flow and readjust or equalize loading. Fasteners under load, for instance, are often required to have a certain amount of ductility so that loads on a joint can be equalized across the fasteners used.

The amount of ductility required for this purpose, however, is probably extremely small. Most authorities agree that 1 to 2 per cent elongation is all that is needed in ordinary service, and that even in large steel sections, 1 to 3 per cent is probably all that can be utilized.[4]

These considerations lead to the question, "Just how important is ductility in design?" The answer, of course, depends entirely on the design itself. Most components are designed in the elastic range of the material; the plastic range is not utilized in normal service and serves merely as a certain conjectural protection from fracture if deformed by an overload.

Many parts will fail functionally if permanently deformed, or at least their operation will be seriously impaired. For this type of part the plastic range has practically no significance; all operation must be in the elastic range of the material. Thus, from the ductility standpoint, a cast iron with a

Relative ductility of the seat in this valve permits the ball to deform it when seating so that a tight seal is provided. Ball and seat are both shock-resistant grades of tungsten carbide, but the Carboloy grade 44A ball is harder than the 190 or 55B used in the seats. The carbide balls and seats are said to outwear hardened steel by a large factor in oil well operations



Photo, courtesy Carboloy Dept., General Electric Co.

comparatively large elastic deflection but little plastic deformation capabilities serves as well as a high-ductility steel with better plastic elongation but considerably less elastic. All other characteristics being equal (which of course they are often not), parts can be designed for brittle materials to serve as well as parts made from ductile materials in which only the elastic range is utilized in normal service. Practical examples of this type of design are camshafts and crankshafts, in which deformation is intolerable; cast iron is often used with completely satisfactory results.

Ductility and Overload Capacity: Elongation and reduction of area are not in themselves measures of capacity to resist overloads; ultimate strength compared with yield strength is the more adequate criterion. The two ductility indicators are, however, indicators of how much deflection or deformation can be tolerated before fracture occurs *under slow, steady loading without excessive stress concentration*. High-speed application of load or fatigue loading are different cases and the material must be evaluated by different tests. Since most actual failures are in fatigue, with a certain percentage of accidental failures occurring by more or less high-speed impact, ductility is of very little significance.

Evaluation of Cold-Workability: So far, the use of ductility properties in estimating cold workability of metals is fairly hypothetical; definite correlation exists but no completely satisfactory parameters for relating the two have yet been found. They may never be; processes and production techniques vary widely between companies.

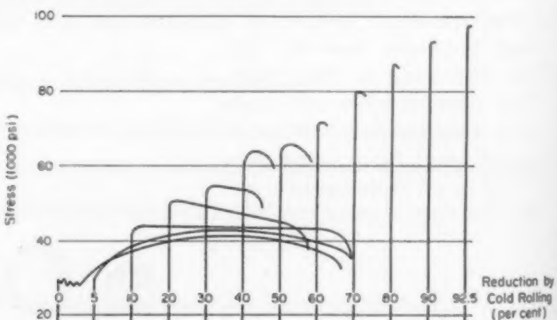
Some similarity has been found to exist between stretch forming and the uniform elongation (not localized necking) in a tension test; a value of 66 per cent of this elongation has been suggested as a generally applicable stretch-forming criterion.[5] Reduction of area has been found to have a close correlation to minimum bend radius. However, reduction of area is difficult to determine on tension test specimens of sheet metal; a useful criterion has been found to be the strain at fracture of a specimen 0.01 to 0.12-inch in length. Deep drawing presents a more complex problem. A suggested criterion is comparison of the ratio of true stress to ultimate stress with true strain, *Fig.* 3.[5] In the critical strain region, the lower the curve, the better the material for deep drawing. This criterion correlates reasonably well for those materials which fail in drawing by fracture at the punch radius.

Cupping tests, such as the Olsen or Erichsen tests in which a punch and die are used to draw a cup in the material until it fractures, are not generally considered to be adequate measures of cold workability, since results vary widely from operator to operator and machine to machine. Most

Initially ductile, the manganese steel in this "whanger" for a waste-paper pulper is hardened and strengthened by continual pounding of foreign material, thus increasing toughness of the metal. The casting is bolted to the outside edge of an agitator situated at the bottom of a pulping vat where it knocks foreign material to the edge. Manganese steel has successfully replaced hard-faced stainless steel in the application

Photo, courtesy American Brake Shoe Co.

Fig. 5 — Effect of cold rolling on the stress-strain curve for ingot iron



specialized tests of this nature are used as in-plant methods for checking consistency of characteristics between different lots of the same metal.

Ductility in Engineering Materials: Representative ranges for elongation and reduction of area for a number of engineering materials are shown in *Fig.* 4. Intended only to show possible ranges, this graph includes values for materials in annealed and heat-treated, soft and cold-worked, states.

Some materials, by virtue of low or high elongation values, do not fall readily into the general range shown. Laminated and most thermosetting plastics, for example, have very little ductility; the stress-strain curve in tension is very close to a straight line with usually less than 1 per cent plastic elongation. In compression, however, plastic deformation may go as high as 5 to 7 per cent.

Cast iron is another material that shows little plastic elongation. Gray iron may have up to 3 per cent plastic elongation; white and mottled irons probably have from 0 to 0.05 per cent. These figures, naturally, do not include complete deflection capabilities of the material, since the elastic deflection range adds an appreciable percentage. Ceramics, glass, carbon and graphite similarly have close to zero plastic range.

By contrast, some thermoplastic plastics, such as nylon, polyethylene, vinyls, etc., have an unusually large combined plastic and elastic strain at fracture ranging up to 400 per cent. Rubber, too, can be formulated to have a large deflection at fracture, although for common elastic varieties, most of this deflection is elastic and not plastic.

TEST VARIATIONS: Elongation of different materials can be compared only if gage lengths are known. Standard tensile-test specimen gage lengths are shown in TABLE 1. Most of the elongation takes place in the "necked-down" area of the specimen; consequently, longer gage lengths will give lower values for elongation. Reduction of area is largely independent of variations in gage length except for short specimens; the customary 2-inch gage-length specimen is slightly shorter than the length beyond which there is no effect. Shorter bars may show lower values. But because of the relative independence from the gage length factor, reduction of area is sometimes considered to be a more reliable figure than elongation.

Shape and size can also influence values obtained, although again reduction of area is less affected. For any comparison at all of elongation values, gage lengths of two specimens must be in the same proportions to their areas. For example, a 4:1 gage length to diameter ratio is necessary in comparing miniature specimens with the standard 0.5-inch diameter, 2-inch gage length tensile specimen.

Reduction in area is relatively unaffected by shape of the specimen. As long as width-to-thickness ratios for square specimens do not exceed approximately 5:1, direct comparison may be made with round cross-section specimens.

Factors Affecting Ductility: As pointed out in Part 2 of this series, November, Page 161, many

Fig. 6—Changes in ductility as affected by section size and notch sharpness for fully annealed silicon-killed SAE 1025 steel

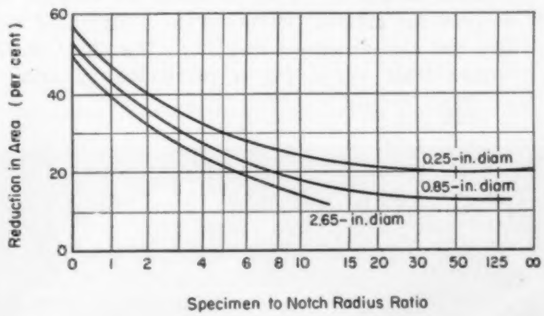


Table 1—Typical Tensile Test Specimens

Type	Applicable Forms	Cross Section	Diameter or Width (in.)	Gage Length (in.)
Plate	Plates, shapes and flats over 3/16-in. thick	Rectangular	1½	8
Sheet	Sheet, plate, flat wire, strip band, hoops, rectangles and shapes from 0.005 to ½-in. thick and over ⅝-in. wide	Rectangular	0.5±0.01	2
Round	General	Round	0.5±0.01	2
Small-size	General, where round specimen too large	Round	Proportional to round	Proportional to round*
Casting	Castings, except malleable	Round	0.505, 0.75, 1.25	..
Malleable	Malleable iron	Round	⅝	2
Die casting	Die castings	Round	0.250	2
Plastic	Molded	Rectangular, 0.125-in. thick	0.5	2

*Not less than 1 in. for copper and alloys.

factors can influence, or be used deliberately to change, the properties of materials. Even supposedly identical lots of the same material will not have identical properties. The main factors are:

1. Heat treatment
2. Cold working
3. Section size and form
4. Mass effect
5. Directionality
6. Variations between lots

Variations in ductility between specimens and within a specimen are always greater than variations in other tensile properties.

HEAT TREATMENT: Heat treatment may be used to soften or harden a metal. Generally, the influence on ductility is related to the amount of hardening and is some inverse function of the tensile strength *for a single material.* Higher tensile strength usually means lower ductility, although a few isolated materials show both higher strength and ductility after a strengthening heat treatment (tungsten-copper-nickel "heavy" metals, for example). The same is not true when comparing two different materials. Alloy steels, for instance, usually have higher tensile strength and ductility than carbon steels of similar carbon content. But tensile strength for a particular material should never be used for evaluation purposes without considering ductility properties and/or hardness, since the three properties are reasonably closely interrelated.

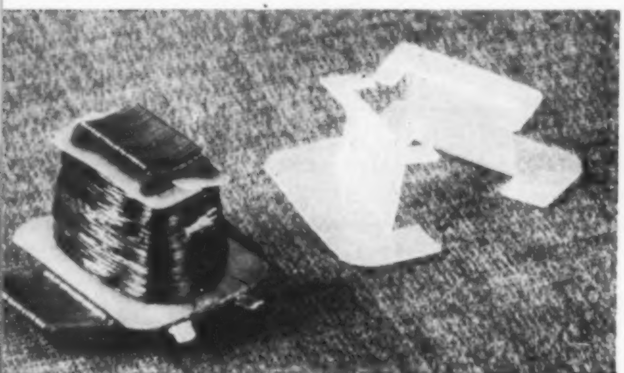
COLD WORKING: Cold working, as pointed out in previous articles, increases the strength of metals, but also decreases ductility. The effect on the stress-strain curve for a particular material is shown generally in *Fig.* 5.[6] Metals show generally similar effects, with a sharp increase in the elastic portion of the curve and a corresponding decrease in plastic strain.

SECTION SIZE AND FORM: Both size and shape of the part affect ductility. Larger section sizes of the same material will generally show lower ductility unless the material has been strengthened by heat treatment. In this case, larger sections may be more ductile because of incomplete hardening through the section.

Shape effects usually show up as areas of stress concentration which, if sufficiently pronounced, may cause the material to break in a "brittle" fashion before full ductility is developed. Combined effects from both causes are shown in *Fig.* 6.[3]

MASS EFFECT: In cast metals, differences in cooling rates for different-size sections affect ductility. Some studies have shown that both strength and ductility decrease in the center of the section as section size increases; values obtained may, however, depend primarily on solidity of the core, which in turn depends on the foundry practice. Bend tests with a notched bar as shown in *Fig.* 7,[7] indicate that larger sections are less ductile, particularly when stress concentration is involved.

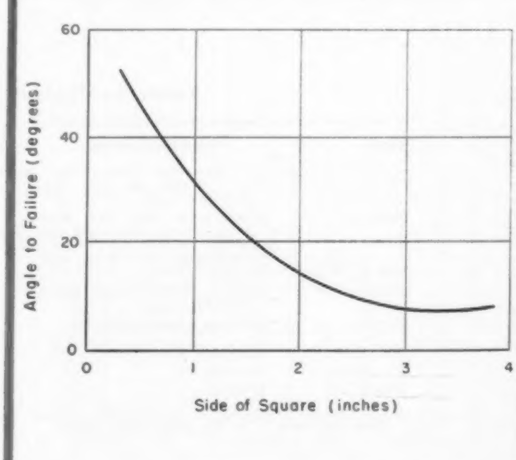
DIRECTIONALITY: Cast metals are isotropic; properties are reasonably uniform in all directions. Wrought metals are not uniform; ductility in the transverse direction—perpendicular to the rolling direction—is likely to be lower than in the longitudinal direction. Ductility in the through-plate direction is usually definitely lower; commercial steel plates with 30 per cent elongation and 65 per cent reduction of area in the longitudinal direc-



Photo, courtesy Polymer Corp. of Pennsylvania

Formability is a prime requisite for these FM-10001 nylon coil forms. Blanked from 0.015 by 1-5/16-inch strip, the nylon must be sufficiently ductile to withstand 90-degree bends without fracturing. The ingenious design eliminates the necessity for an expensive injection mold

Fig. 7—Mass effect on ductility of cast metals; material is mild steel cast in square cross section with a 45 degree V-notch



tion often show only 2 per cent elongation and 5 per cent reduction in the through-plate direction.[8]

Cold reduction has a rather strong effect on directionality of ductility properties; cold-rolled sheet, for instance, often has a definite "grain" which must be considered in designing formed or drawn parts.

VARIATIONS BETWEEN LOTS: As pointed out in previous articles, variations beween different lots of supposedly the same material can change reported property values considerably. Often differences which are not particularly significant in design because of incorporation of definite safety factors, can have a definite effect on ease of processing. Companies using sheet metals for stamping, for instance, often find it necessary to separate lots of the same material into two grades, for severe draws or for milder forming, because of ductility variations between lots.

OTHER EFFECTS: Temperature, corrosion, humidity and many other physical conditions affect ductility of engineering materials; these will be treated more extensively in later articles.

Summary: Ductility figures have very little direct use in design. Elongation and reduction of area, most frequently utilized because of their relative availability as byproducts of a tensile test, are direct measures of the plastic strain capabilities of a material. These ductility values, however, should not be assumed to indicate true fracture-resistance characteristics, since capacity for resisting breaks depends as much on other properties as on ductility. As a rough indicator only, ductility can be used as a very approximate measure of cold workability and resistance to deformation.

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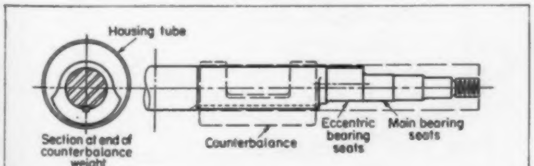
Built-Up Shaft Solves Design Problem

CONVENTIONAL design of an eccentric shaft for a vibrating screening machine, *Fig.* 1, was to turn it from a piece of bar stock of sufficient diameter to include the eccentric throws, and to mount keyed counterbalance weights on the trued, full-size central portion of the shaft. Since a housing tube limited the off-center distance of the balance weights, balance weights had to be quite long requiring machining of long keyways and truing of the shaft for the same distance.

Faced with the problem of producing a shaft with larger bearings for heavier duty, but interchangeable with the previous shaft, it was found that little space was available for counterweights because of the increased shaft diameter and impossibility of an appreciable increase in housing tube size. The problem was solved by using a composite, self-counterbalanced shaft of simple design assembled by welding, *Fig.* 2.

Central portion of the shaft, spanning the distance between the eccentric bearing and containing the counterbalance, is made from standard-weight 8-inch pipe. The counterbalance weight is rough-turned from 8-inch steel bar to 1/16-inch less than the pipe ID, torch-cut in half down its length, cut to length, then welded in the pipe.

Fig. 1—Eccentric shaft was originally turned from 4¾-inch diameter bar stock. Counterbalance weights were cast and keyed to the shaft



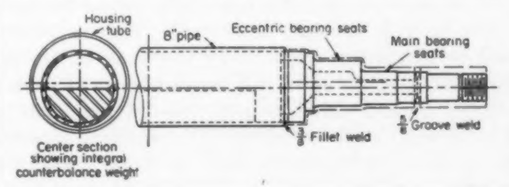
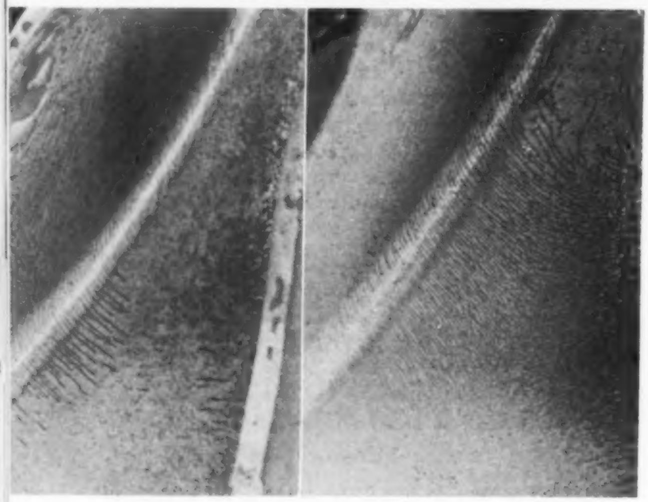


Fig. 2—Welded-in, half-round counterbalance weights and built-up construction make heavier bearing seats possible with little increase in housing tube diameter

Bearing seats are cast of 0.40 per cent carbon steel to provide added hardness. Shaft end extensions are machined from C1040 steel rather than being cast integrally with the bearing seats since the long extension at the small end of the casting might have presented some difficulties in the foundry. Bearing seat castings and shaft extensions are joined to the center portion of the shaft by electric arc welding. Component pieces are cut as necessary for fillet or groove welding.

From a paper receiving Honorable Mention in a Mechanical Design Award Program of the James F. Lincoln Arc Welding Foundation by David J. Klyce, engineer, Symons Brothers Co.

Fig. 1—Brittle coating crack patterns for two successive rotor "spin" tests



Ceramic Stress Coating

. . . facilitates experimental analysis of rotating parts at high speeds and elevated temperatures

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ANALYSIS of stress distribution in rotating bodies becomes a particularly vexing problem at high speeds and elevated operating temperatures. In the development of Solar Aircraft's Mars radial flow gas turbine, this problem was accentuated by a rated operating speed of 40,000 rpm and an inlet gas temperature of 1350 F. Design of the rotor wheel necessitated experimental verification of theoretical calculations to assure reliability of stress data.

Solution to the problem was finally found through a new fired-on ceramic base brittle coating developed by Magnaflux Corp. under the name of Stresscoat All-Temp. Similar in performance to the more familiar lacquer coatings, the new coating consists of a carefully selected ceramic slip which is sprayed on the part to be tested, fired at 1080 F and air cooled.

Use of the ceramic base coating is not affected by oils, humidity, most corrosive substances or by temperatures up to around 700 F. Since the Mars turbine rotor operates at temperatures well above 700 F, all loading and testing was done at room temperature but this did not nullify the usefulness of the results.

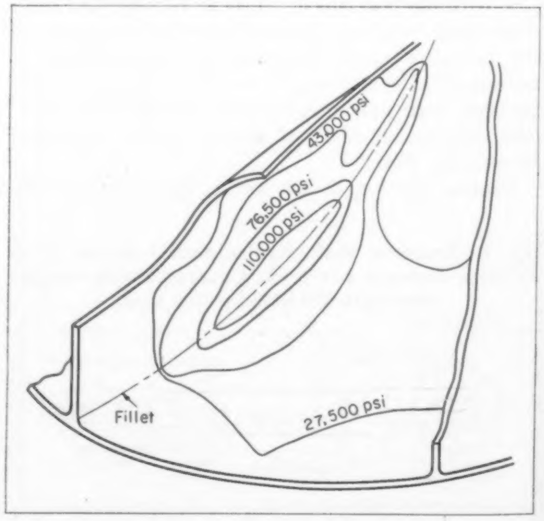
In brief the testing procedure was accomplished as follows:

1. Proper slip was selected and applied to two rotors and two calibration bars.
2. Strain threshold was found from calibration bars.
3. Rotors under test were spun in an evacuated pit at successive speeds from 21,000 to 50,000 rpm.
4. A special powder was sprayed on cracks to make strain lines visible and photographs were taken for permanent records, *Fig.* 1.

At 25,000 rpm, the first strain lines appeared in about the middle third of vane fillet, indicating a stress of 27,500 psi. Assuming the stressed material to be within the elastic limit, and that stress varies as the square of the ratio of speed, a series of stress areas or boundaries of equal strain (isoentatics) can be obtained. At each increasing speed the value of stress in this area is corrected and so is each succeeding area. The final result is a series of contour lines within which the stress is known to equal or exceed a calculable amount, *Fig.* 2.

Strain lines on the reverse side of the rotor were so entirely different from the pattern on the vane side that it was obvious that the vane loading of the disk was the cause of the critically stressed areas. Since the stress causing the strain is at right angles to the strain line, stress direction at any point can be found by inspection of the crack patterns. Data thus obtained substantiated the belief that the vanes were the major cause of disk stress and, at the same time, gave definite values of stress as well as locating areas of concentration.

Fig. 2—Final pattern of isoentatics for rotor of *Fig.* 1 showing stress areas at 50,000 rpm which were determined from successive spins of the rotor



*Now with Department of Mechanical Engineering, California State Polytechnic College, San Luis Obispo, Calif.

CHART SIMPLIFIES

Design of Gear Pumps

By Reiner J. Auman
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IN THE design of gear pumps, quantity of liquid discharged in unit time is of prime importance. Of course, this delivery factor is determined by pump size, speed, and efficiency. To calculate the delivery or output of a gear pump it is first necessary to find the volumetric displacement of the particular gears employed in it. For standard involute gears, *Fig.* 1, this computation can be made from the familiar approximate displacement formula

$$Q = \frac{nF}{231}\left(\frac{\pi\, d_o^2}{4} - \frac{\pi\, d_i^2}{4}\right) \quad (1)$$

Since by definition $d_o = D + (2/P)$ and $d_i = D - (2/P)$ for full-depth involute gears, *Fig.* 1, Equation 1 reduces to

$$Q = \frac{nF}{231}\left(\frac{\pi}{4}\right)\left(\frac{8\,D}{P}\right)$$

$$= 0.0272\,\frac{nFD}{P} \quad (2)$$

Also since $D = N/P$, Equation 2 can be expressed as

$$Q = 0.0272\,\frac{nFN}{P^2} \quad (3)$$

Equations 2 and 3 can be employed as the bases for several relationships based upon principles of homology. That is, from Equation 2

$$\frac{Q_1}{Q_2} = \frac{n_1F_1D_1P_2}{n_2F_2D_2P_1} \quad (4)$$

and from Equation 3

$$\frac{Q_1}{Q_2} = \frac{n_1F_1N_1P_2^2}{n_2F_2N_2P_1^2} \quad (5)$$

These relationships permit the computation of actual delivery of one pump from known characteristics of another. The only "theoretical" error is that arising from the difference in efficiency between the two pumps. Furthermore, if certain of the variables are the same for the two pumps being compared, several basic relationships become apparent. For example, if $n_1 = n_2$, $F_1 = F_2$, and $N_1 = N_2$, then from Equation 5

$$\frac{Q_1}{Q_2} = \frac{P_2^2}{P_1^2} \quad (6)$$

Such simplified statements of proportion holding

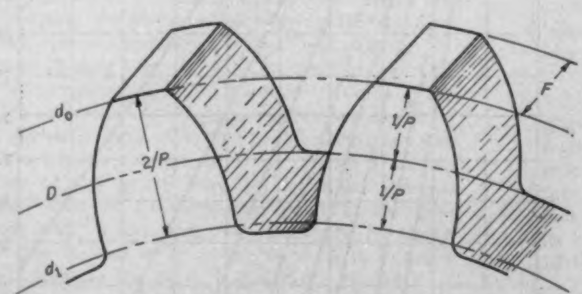


Fig. 1—Basic dimensions for standard involute spur gear

Nomenclature

- D = Pitch diameter, in.
- d_i = Inside diameter, in.
- d_o = Outside diameter, in.
- F = Face width, in.
- N = Number of teeth
- n = Rotational speed, rpm
- P = Diametral pitch, teeth per in.
- Q = Quantity of liquid discharged in unit time, gpm
- η = Efficiency of pump, per cent

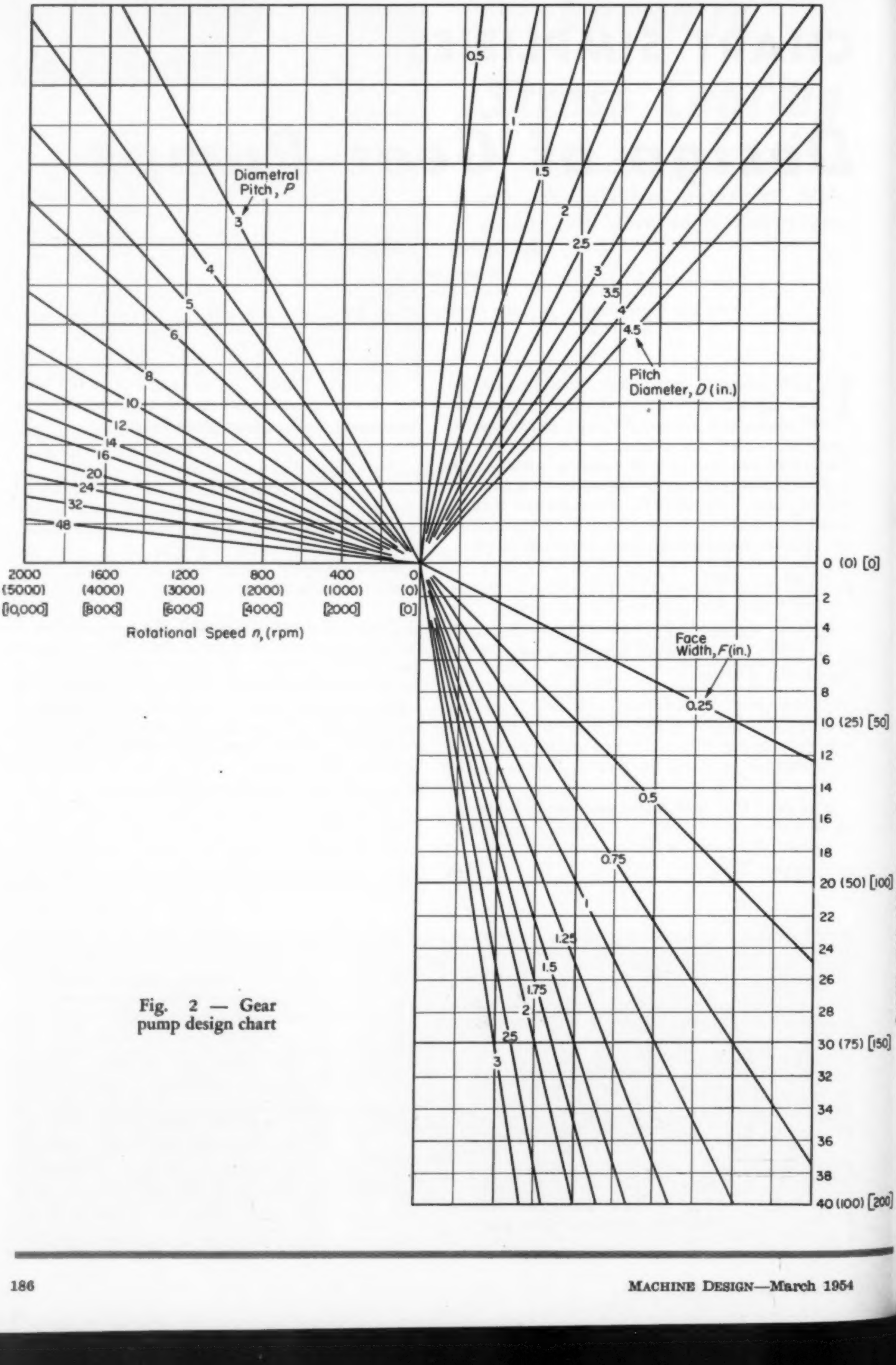


Fig. 2 — Gear pump design chart

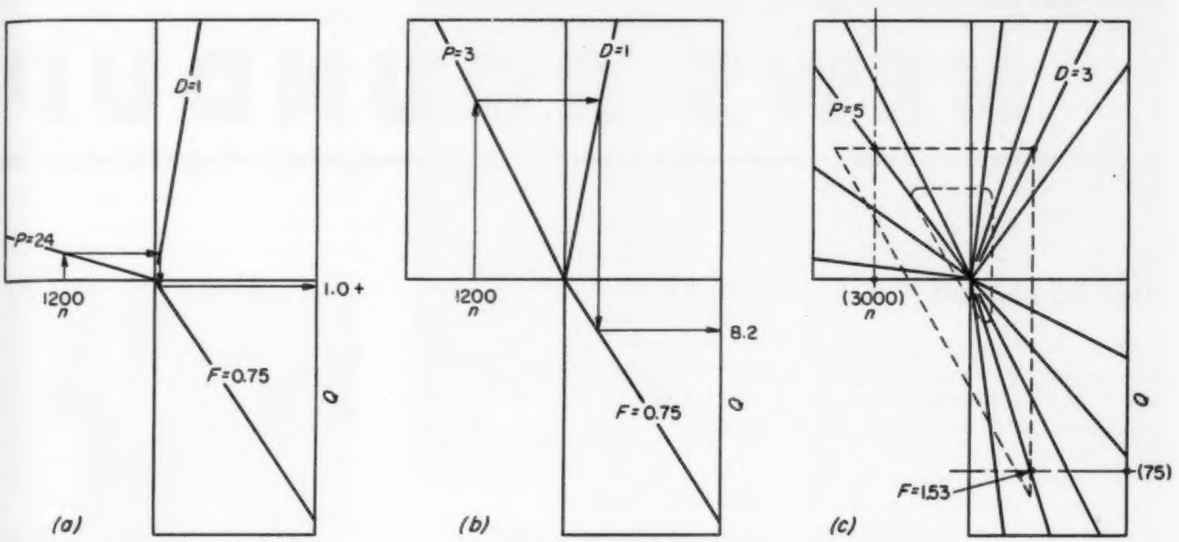


Fig. 3—Chart keys for examples: *(a)* and *(b)*, alternate solutions for Example 1; *(c)*, solution for Example 2

between liquid output in unit time and dimensions or speed are frequently useful for rapid estimation in design.

Whenever original design is initiated, a quick graphical method for determination of gear pump dimensions, speed and output is provided by the chart shown in *Fig.* 2 which is plotted from Equation 2. Use of the chart naturally depends upon given or suitable pump dimensions, delivery and speed; of course, many of these factors are somewhat optional. The following examples point up the utility of this chart.

EXAMPLE 1. For a gear pump having a speed $n = 1200$ rpm and gear dimensions of $P = 24$, $D = 1$ inch, and $F = 0.75$-inch, what is the output capacity Q in gpm? Alternate methods of solution can be applied with the chart in *Fig.* 2.

Solution A (*Fig.* 3*a*). Start at the 1200-rpm point in the gear speed scale and rise vertically until the $P = 24$ line is intersected. From this intersection go horizontally right until the $D = 1$ inch line is crossed, then drop vertically to the $F = 0.75$-inch line. At this intersection go horizontally right until the quantity scale is intersected and read $Q = 1$ gpm, approximately.

Solution B (*Fig.* 3*b*). For greater accuracy, use of the curves in regions more remote from the center of the graph may be desirable. For example, start at the 1200-rpm point in the gear speed scale again and rise vertically to the $P = 3$ line instead of $P = 24$. Then as in Solution *A*, go right to the $D = 1$ line, down to the $F = 0.75$ line, and right to 8.2 gpm on the quantity scale. Changing $P = 24$ to $P = 3$ has the effect of multiplying Equation 2 by 24/3 or 8; therefore, the answer is $Q = 8.2/8 = 1.02$ gpm, approximately.

Since gear pump efficiency usually enters into most actual calculations, the following example shows how to handle this factor.

EXAMPLE 2. If the anticipated pump efficiency, η, is 80 per cent, required liquid output $Q = 60$ gpm, and rotational speed $n = 3000$ rpm, what would be a suitable gear diametral pitch P, pitch diameter, D, face width F, and number of teeth N?

Since the pump will be only 80 per cent efficient, design output capacity is Q/η or $60(100)/80 = 75$ gpm. On *Fig.* 2, lay off a horizontal line through $Q = (75)$ gpm and extend this line across the gear width lines. Lay off a second line vertically upward from $n = (3000)$ rpm which cuts across all diametral pitch lines. By using a right triangle, keeping the two legs parallel to principal axes of chart, *Fig.* 3*c*, position the triangle so that the apex of the right angle lies within the range of pitch diameter lines while the two triangle legs intersect the previously laid out lines within the range of diametral pitch lines and gear width lines. By moving and scanning the triangle, diametral pitch, pitch diameter, and face width can be selected arbitrarily to meet any specific design limitations or requirements. Therefore, one of many practical solutions to this problem is $P = 5$, $D = 3$ inches, and $F = 1.53$ inches, as indicated in *Fig.* 3*c*, and $N = DP = 3\ (5) = 15$ teeth.

Integers for number of teeth and diametral pitch must, of course, be kept in mind.

Engineering
NEWS ROUNDUP

Atom Sub Launched

U. S. S. Nautilus, the world's first atomic powered submarine, was recently launched at the Groton, Conn., yards of Electric Boat Div., General Dynamics Corp. Said to be capable of crossing the Atlantic Ocean submerged at speeds in excess of 20 knots, the Nautilus with its atomic engine, developed by Westinghouse Electric Corp., is an outstanding example of the application of nuclear fission to the production of useful power.

Although military necessity has resulted in production of this vessel despite present costs of reactors, lessons learned with this power plant should hasten the day of economical atom power. Power developed by the reactor is said to make the Nautilus the most powerful submarine ever built. Fleet type submarines of World War II had engines of about 6000 horsepower.

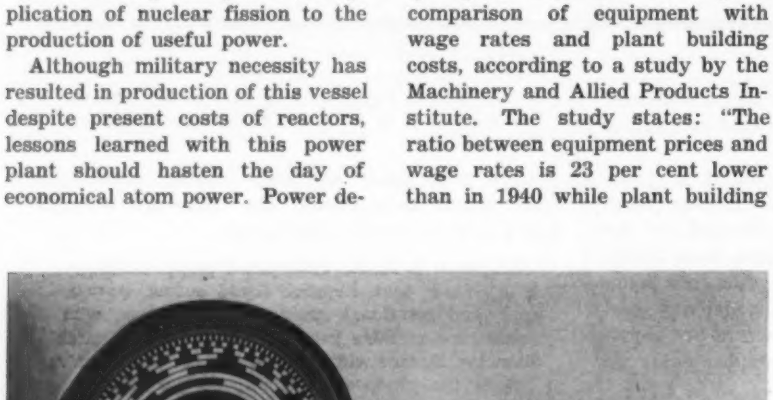
Machine Prices Increase Mechanization

Productive equipment is currently a good "buy" in terms of price comparison of equipment with wage rates and plant building costs, according to a study by the Machinery and Allied Products Institute. The study states: "The ratio between equipment prices and wage rates is 23 per cent lower than in 1940 while plant building costs are currently as high relative to wage rates as in prewar years."

Because of steadily increased productivity in the industries producing machinery and equipment, prices have moved down steadily relative to wage rates during the last century. The Institute charts this downward movement since 1900, reporting "Not only has there been a reduction in the equipment price-wage rate relationship since 1940, but the current ratio between the two is somewhat lower than a projection of the long-term trend would indicate."

"Between 1900 and 1940 the re-

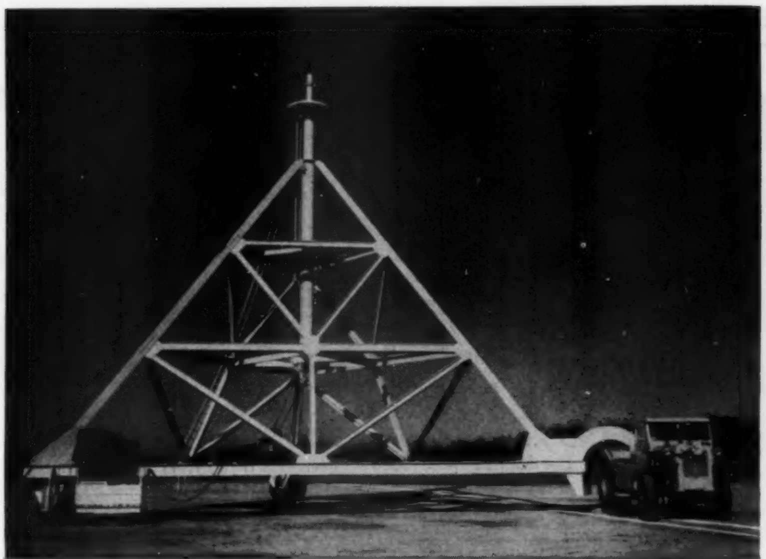
LONG AND SHORT of it are these forged stainless steel turbine blades. Manufactured by the Kropp Forge Co., the small blade in the workman's hand weighs 1/4 pound and is used in jet engine compressors. Large blade, weighing 1100 pounds and almost 5 feet high, is for the huge aircraft testing wind tunnel at Tullahoma, Tenn.



CODE WHEEL for digital computer is 5½ inches in diameter and has 4096 alternating opaque and transparent sectors in the outer of 13 concentric rings. Outer ring sectors are less than 0.1-inch long and 0.004-inch wide. Made from glass by a photographic method, the wheel is a Bell Telephone Laboratories design produced by W. & L. E. Gurley. Light, passing through the wheel and a slit to phototransistors, converts analog to binary digital data

lationship between plant building costs and wage rates also moved steadily lower, at about the same rate as the equipment ratio. Since that time, however, the picture has changed considerably." According to the Institute study, "The plant ratio in 1953 was as high as in 1940 and some 24 per cent above trend. Thus, while equipment prices have been reduced significantly relative to wage rates in the past decade, plant costs have remained about as expensive as labor costs."

The decline of equipment prices relative to labor and plant is held to be significant both as an explanation of postwar industry trend toward increased mechanization and as an indication this trend will continue in the future.



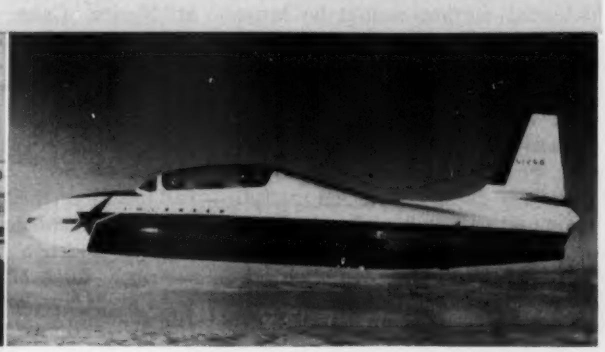
MOBILE MOORING MAST for ground handling of huge nonrigid airships of 975,000 cubic foot size has been improved by mounting one corner directly on the towing tractor. New design eliminates the fifth wheel and towing tongue previously used by substituting an Athey draw-bar gooseneck. The joint development of Goodyear Aircraft Corp. and the U. S. Navy decreases rolling resistance and turning radius, and permits use of more powerful and faster tractor because of increased traction of the tractor wheels. Tractor is a Caterpillar Model DW-10

Inventors Have Their Day

Inventors will have an opportunity to make a bid for profitable recognition of their "brain children" at the Invention Exhibit and Conference to be held May 3 at the Cleveland Engineering Society. Inventors having a practical product or process are invited to submit patent papers, or resume, and a commercial description, if available, to the Invention Conference Group of the Cleveland Engineering Society, 2136 E. 19th St., Cleveland 15, O., not later than April 1. Nonpatented items are not acceptable for consideration unless submitted with a disclosure form obtainable from the Society. Inventors are cautioned not to send original patent papers since all material submitted will not be returned but will become the property of the Society to be deposited in its library. There are no entry fees connected with the exhibit.

In judging entries the conference

NEW JET TRAINERS are the North American TF-86, left, and a two-place Lockheed design, right. Basically an F-86F Sabre jet, the TF-86 has been lengthened 5 feet to accommodate tandem cockpits and dual controls. Speed of the TF-86 is in excess of 650 mph, service ceiling is 45,000 feet, and operating radius is 600 miles. Lockheed's trainer retains many features of the standard T-33 jet trainer used by the U. S. Air Force and also incorporates such new ideas as a raised rear seat for the instructor, to provide better visibility for observing the students' actions as well as for flying. Speed of the Lockheed trainer is 600 mph wing span is 42 feet.



WILLYS AND KAISER 1954 models have been announced by the Kaiser-Willys Sales Div., Willys Motors Inc. This report concludes MACHINE DESIGN'S coverage of the 1954 cars started in the January and February issues. Both Kaiser and Willys have increased horsepower. Although basic bodies are the same, the Kaiser hood, grill and headlights have been changed radically. There should be no difficulty distinguishing the 1954 model from earlier Kaisers. A supercharger has been fitted to the L-head, in-line, 6-cylinder engine of the Kaiser Manhattan to boost horsepower to 140. Horsepower per cubic inch of displacement is 0.619, a quite respectable figure exceeded by some, but not by all of the current overhead-valve, V-8 engines. Power steering, power brakes and Hydra-Matic transmission are available as extras on the Kaiser. Willys Eagle and Ace models have a new 115-horsepower L-lead, 6-cylinder engine and last year's 90-horsepower F-head has been handed-down to the Lark. New alloy intake and exhaust valves with rotators are used in all engines. Front suspension and steering have been improved. A variety of new exterior colors and interior fabrics are used. Windshield wipers have been designed so their strokes overlap to clean the center of the one piece windshield. Power steering and Hydra-Matic transmission are available as optional equipment

Engine Specifications

	Willys Lark	Willys Ace, Eagle	Kaiser Special	Kaiser Manhattan
Type	F-head	L-head	L-head	L-head
No. Cyls.	6	6	6	6
Bore & stroke (in.)	3.125 × 3.5	3.313 × 4.375	3.313 × 4.375	3.313 × 4.375
Displ. (in.^{3})	161.0	226.2	226.2	226.2
Comp. ratio	7.6 to 1	7.3 to 1	7.3 to 1	7.3 to 1
Bhp, max	90 @ 4200 rpm	115 @ 3650 rpm	118 @ 3650 rpm	140 @ 3900 rpm
Torque, max (lb-ft)	135 @ 1600 rpm	190 @ 1800 rpm	200 @ 1800 rpm	215 @ 2400 rpm

Size and Weight

	Willys	Kaiser Special	Kaiser Manhattan
Wheelbase (in.)	108	118.5	118.5
Length (in.)	180¾	213.78	215.62
Width (in.)	72	74.875	74.875
Height (in.)	69.75	60.25	60.25
Shipping Weight (lb)	2778*	3210	3275

*Ace-Eagle, 2847—Lark, 2661.

group will use the following as a yardstick of a "practical product or process". The patent should fill a specific human need; should be better than other comparable products or processes; should have its research development completed so that it is ready, or almost ready to be put into production; should preferably be one which can be readily sold through existing sales or merchandise channels; and the potential market should be large enough in size and the difference between the cost and the sales price should be great enough so that substantial profits may be expected. The conference group will also be interested in knowing if the item is available as a full scale unit or as a model and whether it is operating or nonoperating.

In addition to the exhibit of selected inventions, ideas and processes, there will be a program covering such phases as "Technique of Invention," "Fitting an Idea into Production," "Fitting an Idea into Marketing," "The Future of Invention in Industry," and "The Manufacturer Looks to Invention."

Nickel Supply Will Be Increased

A government owned nickel plant at Nicaro, Cuba, now producing at a rate of about 28 million pounds per year, is to be expanded in capacity by 75 per cent. In announcing the expansion, the General Services Administration also announced that the project had been certified as being essential to national defense by the Office of Defense Mobilization. A research fund of $1 million has been made available in addition to the $43 million for expansion. Research on improved processes for refining nickel ores will be conducted.

The General Services Administration is also negotiating projected construction of a pilot plant at Moa Bay, Cuba, with the Freeport Sulphur Co. Pilot plant production would be 50 tons per day of nickel and cobalt by a new chemical process.

Pennsylvania State University will have the second nuclear reactor authorized by the AEC for an educational institution. The low power "swimming-pool" type reactor will be operated for nuclear research and for training students. Operating at 100 kilowatts, the reactor will be cooled and moderated with ordinary water and will use enriched uranium fuel. Fuel ele-

Design for economy with JOHNSON BEARINGS

IN THE VAST MAJORITY of applications Johnson Sleeve Bearings give long, troublefree service. Being one-piece bearings and precision made, they are economical in installation. Furthermore, they are low in first cost. Johnson Bearings are available in a wide range of bearing metals, alloys and combinations: aluminum on steel; bronze on steel; babbitt on steel or bronze; powder metallurgy; cast bronze; cast aluminum alloy; and sheet bronze, plain or graphited. No other manufacturer can furnish all of these types . . . that is why Johnson Bronze is known as "Sleeve Bearing Headquarters." Plan to save money with Johnson Bearings. Free engineering consultation is available. Write.

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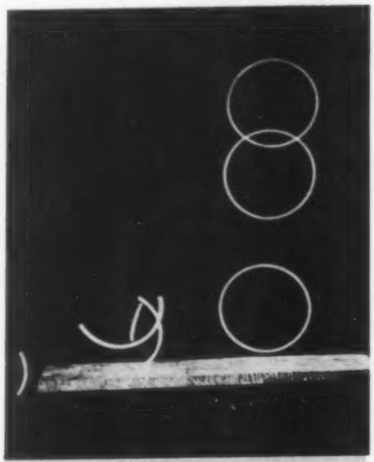

BOUNCING RING in this stroboscopic photo is a section cut from cast ductile-iron pipe and dropped 20 feet onto a steel plate. What happens to a similar section of ordinary cast iron is shown at the left. This International Nickel Co. photo demonstrates graphically the shock resistance of ductile iron

ments will be suspended in a pool of water sufficiently deep to serve as a shield against radiation produced by the reactor. The project is under the direction of Dr. William M. Breazale, professor of electrical engineering.

Functions of business advisory and technical committees assisting the General Services Administration in drafting specifications and standards used in Government procurement operations have been officially outlined in a regulation issued recently by Edmund F. Mansure, Administrator of General Services.

Under certain conditions, technical specialists from technical societies and trade associations may be invited to furnish specialized knowledge and the benefit of their experience on specific projects related to Federal procurement matters. Principal difference between the business advisory committees and technical committees is that the latter will not examine policy matters but will be concerned only with developing engineering and other technical data used in purchase specifications and standards. Further information may be obtained from Mr. Mansure at the General Services Administration, Washington 25, D. C.

Industry Admits Equipment is Obsolete

By industry's own admission, 30 per cent of the machinery and processes, 28 per cent of automation equipment and processes, 34 per cent of inspection methods, 28 per cent of metal forming processes, 23 per cent of grinding and finishing equipment, and 25 per cent of production welding equipment are obsolete. These figures, based on a recent survey made by the American Society of Tool Engineers, have been published by the ASTE in a pamphlet entitled "Are YOU Ready to Meet Stiff Competition?"

Tooling for competition is the keynote of the 1954 Industrial Exposition and General Meeting of the ASTE, to be held in Philadelphia April 26 through 30. The booklet also tells how the Exposition has been designed to aid industry in preparing for competition.

Steelmaking Capacity Increased in 1953

An increase of about 6.8 million tons in steelmaking capacity took place in the United States during 1953 according to figures recently released by the American Iron and Steel Institute. Total capacity is now 124.3 million tons. The increase occurred in 14 states of the 27 listed by the AISI as steel producers. Some decline in capacity occurred in three states.

Largest gain, 1,744,800 tons, was in Ohio. Second largest increase was made in Indiana, 1,276,500 tons. Other major increases were: Illinois, 1,079,600 tons; Michigan, 744,880 tons; Pennsylvania, 685,-900 tons; Texas, 520,180 tons; Alabama, 376,000 tons; Maryland, 350,000 tons; New York, 254,830 tons; and Utah, 204,000 tons. Five largest steel producers are Pennsylvania, Ohio, Indiana, Illinois and Michigan, in that order.

The second Human Engineering Institute will be held in Stamford, Conn. the week of May 10. Spon-

ENGINEERING CENTER for use by local sections of national en-engineering organizations will be built in downtown Cleveland, O., to replace the present Cleveland Engineering Society building. To be known as the Cleveland Engineering Center, the two-story structure will have an auditorium seating 900 persons, dining rooms for 400, and meeting, conference and class rooms. Off-street parking space will be provided

(Advertisement)

Make Office Photocopies in Brighter Light on Peerless DRI-STAT *"Bright-Light" Paper*

NEW DRI-STAT "Bright-Light" paper offers, for the first time, office photocopying that truly deserves the name. Developed by Peerless Photo Products, Inc., DRI-STAT "Bright-Light" paper allows you to make sharp, clear, black-and-white photocopies of any original, under the normal levels of illumination in today's modern offices . . . fluorescent or incandescent light of fairly high intensity, or subdued daylight. Here, at last, is a paper that will give you good results under four to five times more light than you could ever work in before.

Now you can really make full use of transfer-process photocopying . . . the modern, quick, inexpensive method of making exact copies of anything written, drawn, or printed . . . anywhere in your office. No longer need you put up with the inconvenience of having to locate your photocopy equipment in an out-of-the-way corner, a stuffy closet or a crowded stockroom. When you use the new DRI-STAT "Bright-Light" paper, you can put your photocopy equipment right where you want it . . . right where it's most convenient and comfortable to use . . and get sparkling, clear, black-and-white copies, regardless of room illumination.

DRI-STAT papers give these amazing results not only on DRI-STAT equipment, but with almost every other type of transfer-process photocopying equipment now on the market. If you already have such equipment, try DRI-STAT papers and see the improvement!

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You can copy originals up to 11″ by 17″ in size on DRI-STAT equipment, or continuous strip matter up to 12″ wide. Results are equally good from opaque or translucent originals, since copying can be done either by the reflex or the contact method.

DRI-STAT vellum paper can be used to produce a translucent "master" from which multiple copies can be reproduced by the diazo process.

DRI-STAT equipment takes up less space than a typewriter, can be used right in the department (or branch office) where photocopies are needed. The originals never leave your possession. Copies are ready in less than one minute, cost less than 10 cents apiece.

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DREAM CARS: Pontiac's Bonneville Special, left, Strato Streak, center, and Buick's Wildcat, right, have at least two things in common. They are not for sale and they have laminated glass fiber and plastic bodies. Bonneville Special is Pontiac's idea of a competition type sports car; height is 48½ inches, length is 158.3 inches and wheelbase is 100 inches. The transparent plastic top is hinged at the center and raises at each side on counter-balanced springs to permit access to the interior door handles. Called a spectator sports car, the Strato Streak has an extremely large glass area and four bucket type seats; front seats pivot 90 degrees to permit easy entry and exit. Despite four-door styling no center post is used on the Strato Streak to allow greater glass area. Height is 54.5 inches, length is 214.3 inches and wheelbase is 124 inches. Buick's Wildcat on a 100-inch wheelbase is powered by a 220-hp engine and displays some radical styling which includes headlamps mounted at the sides of the cowl and chromed, exposed under-part of fenders

sored by Dunlap and Associates, Inc. the five day course is organized to meet the needs of engineers, designers, product planners, safety specialists and members of related professions interested in the basic knowledge of human engineering. Further information may be obtained from Dr. Bernard J. Covner, Director, Human Engineering Institute, Dunlap and Associates, Inc., 429 Atlantic St., Stamford, Conn.

Hot Extrusion Speeds Turbine Production

Aluminum wheels and fans with diameters from 2 inches to 4.5 inches and weights of 2 ounces to 1½ pounds are being extruded hot to finished sizes and difficult contours with tolerances as low as 0.001-inch at the rate of 100 per hour. Made with a specially designed 1100-ton Lake Erie Press at the Phoenix Div. of The Garrett Corp., wheels are used in air cooling, refrigeration and ram air turbines. Fans with ten deep blades, bucket wheels with 30 precision buckets and radial air foil sections all to finished sizes are some of the parts made from tough aluminum alloys, such as 14S.

Since hand duplicating took four hours per fan, the 100 per hour rate represents a considerable saving. In addition to the economic advantage, the quality of the products is far superior since grain flow is radial to the blade ends.

Aluminum slugs are preheated in an oven, placed in the press die with long-handle tongs and extruded in a matter of seconds. Die temperatures are controlled by five heaters in the die itself. Insulation pads prevented the die heat from affecting the press. Once cooled the fans are either heat treated or machined directly, depending upon the alloy. Machining consists of work only on the hub bore or outer diameter; no contour machining is necessary.

Production of polyethylene, one of three critical materials still in short supply (others include nickel and some steel items), has been increased 45 per cent with the full-scale operation of the new Bakelite plant in Texas City, Texas. The increased production is approximately equal to the entire industry's polyethylene production a year ago, announced George C. Miller, president of Bakelite. By mid-1954 expected production will be 120 million pounds per year from this and one other new Bakelite plant now being completed — greatly easing the present shortage.

Commercially Produced Vacuum Melted Metals

First commercial production of vacuum melted metals was recently announced. Vacuum melting permits precise control of very dilute alloying additions and trace impurities to achieve superior high-temperature alloys. Vacuum Metals Corp., producer of the alloys, says that application of these metals to jet engines gives promise of increased ease of fabrication and higher yields as well as superior high temperature performance.

Bearings Co. of America, located in Lancaster, Pa., has been acquired by Federal-Mogul Corp. of Detroit. Initially, Bearings Co. of America will be operated as a division of Federal-Mogul with its service division also co-ordinated.

LOOK TO THE LEADER . . . FOR LUBRICATING DEVICES

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40 YEARS' LEADERSHIP

Yes, for 40 years GITS has been setting the standard for industry . . . solving tough lubricating problems . . . earning the confidence of manufacturers . . . it's the reason people say, "Call GITS first".

LOW COST

Yes, GITS oil cups can do a complete lubricating job for you . . . prolonging bearing life, reducing maintenance costs, cutting down-time, boosting production . . . and GITS oil cups cost so little.

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Yes, only GITS can offer you such a wide range of standard stock sizes. From *just one* source you can get *all* lubrication devices in *any design* for *any purpose.*

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Yes, GITS is known for uniform quality in design, materials and machining . . . this means constant, dependable performance for you. Inferior products can cost you time and money. Demand the best . . . get GITS.

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Write today for Free Catalog No. 60A. Use it as your handy reference for lubricating devices.

J. W. Brady, former Bearings Co. president, was elected a director and vice president of Federal-Mogul and will be in charge of the new division.

Consolidation of two of the oldest bearings companies gives Federal-Mogul a complete line of anti-friction bearings, consisting of thrust and ball bearings, tapered and straight roller bearings, as well as its regular line of sleeve bearings, thrust washers and bushings.

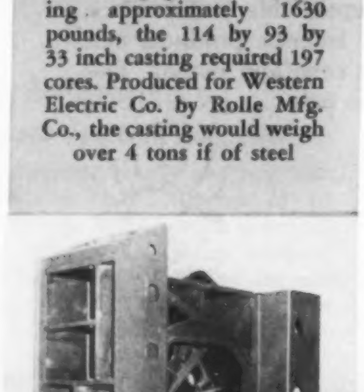
UNDERGROUND LOCOMOTIVE is said to be the largest and most powerful of its type. Rated drawbar pull is 25,000 pounds and the locomotive is capable of pulling 1600 tons on the level. Length is 35 feet, and four 150-horsepower motors are used. Built by GE's Car and Equipment Dept., the locomotive is used by the Mathies Coal Co.

Cast Nickel Alloy Conserves Scarce Metal

A cast nickel alloy, designated as the HF grade, containing approximately 20 per cent chromium and 9 per cent nickel can effectively substitute for an alloy containing about 26 per cent chromium and 12 per cent nickel—HH grade—in the temperature range of 1200 to 1600 F. This information is the result of research performed by Battelle Memorial Institute for the Alloy Casting Institute.

Investigating a broad series of alloys, the Battelle group found the most useful group are the HF alloys, containing 19 to 23 per cent chromium, 9 to 12 per cent nickel and 0.25 to 0.40 per cent carbon. In general the mechanical properties of the austenitic HF compositions are comparable to those of higher-nickel alloys in the 1200 to 1600 F range. At 1400 F, the 100-hour rupture stress of a typical HF composition is about 14,000 psi. This is equal to the representative value for the higher-nickel HH grade. Ductility of HF alloys at room temperatures is superior to that of the HH grade; therefore, the HF alloy not only saves nickel but has certain superior properties for intermediate service.

LARGEST MAGNESIUM CASTING ever poured is this base for military electronic equipment. Weighing approximately 1630 pounds, the 114 by 93 by 33 inch casting required 197 cores. Produced for Western Electric Co. by Rolle Mfg. Co., the casting would weigh over 4 tons if of steel



A Special Products Div. for development of special instruments and instrument components to contract specifications has been announced by Fischer & Porter Co., Hatboro, Pa. Special projects will be handled in developing instrumentation for difficult applications relating to point-of-measurement or remote indication, recording, and/or control of such variables as flow, pressure, temperature, level, specific gravity, viscosity, consistency and others. The company will undertake development of instrumentation for precise measurement, transmission, and control of variables in mechanical, magnetic, pneumatic, electrical and electronic systems.

Computers at Low Cost

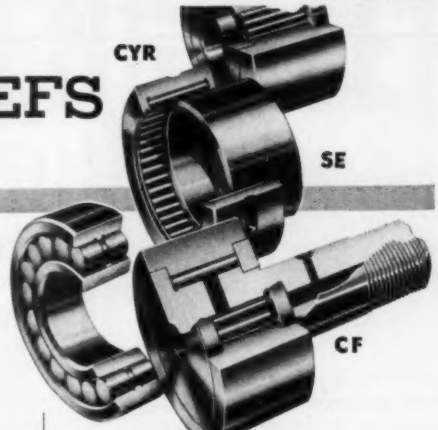
A new electronic computer is said by its manufacturers to do the work of a $1 million machine. Cost is $48,000. Developers of the machine, known as ALWAC, are Logistics Research Inc. President of the firm is Glenn E. Hagen, formerly a research physicist with Northrop Aircraft, supervisor of a computer research group at Northrop and general manager of the Computer Div. of Bendix Aviation Corp.

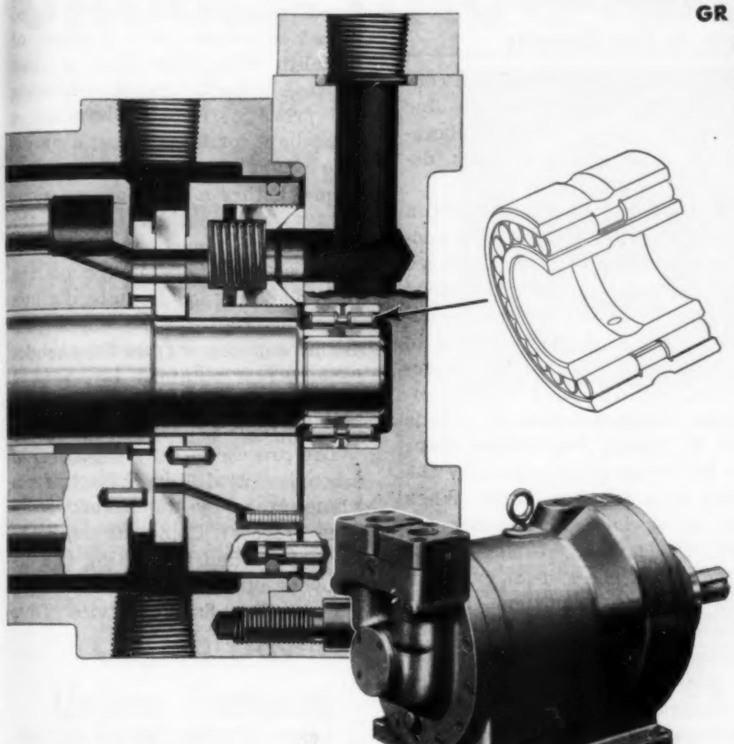
A serial, binary computer with internally programmed magnetic drum, the machine consists of an arithmetic unit, memory unit, control unit and input-output sections. A magnetic drum memory of 2048 word main storage capacity stores numbers and instructions for this machine. Input-output devices may be electric typewriters, paper tape perforators and readers, or automatic graph following and plotting equipment.

Three cabinets on ball bearing

(Advertisement)

McGILL BEARING BRIEFS





GUIDEROL BEARING

used in pilot model Oilgear Motor

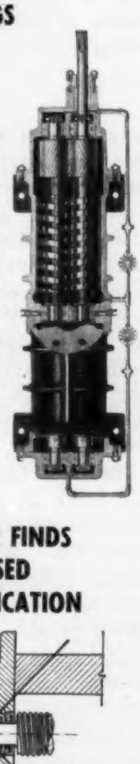
SHOWS NO WEAR AFTER 4000 HOUR TEST

The Oilgear Company, Milwaukee, Wisconsin, a regular McGill bearing user, has been testing a pilot model of a new 60 HP Axial Type Hydraulic Motor equipped with Guiderol Bearings for over a year. After 4000 hours of full load operation, the Guiderol Bearing used on the pilot end of the motor shaft was removed and inspected. It was still in perfect condition showing no internal wear.

This new line of lighter and faster operating axial piston units take advantage of the Guiderol bearing's assured freedom from skewing and extra load capacity in reduced radial bearing space.

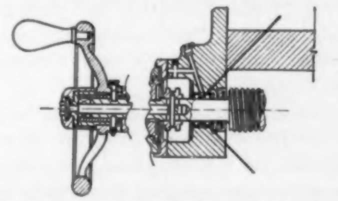
Mr. Ernst Wiedmann, Oilgear Chief Engineer, is enthusiastic about the performance of the Guiderol Bearings. He says: "They are the best full-type roller bearings for our hydraulic pumps and motors." As a result he is changing old specifications for this type of bearing to Guiderol and including Guiderol Bearings on all new developments.

WARREN-QUIMBY ADOPTS GR BEARINGS FOR MAXIMUM RADIAL LOADS



Warren Steam Pump Company has adopted Guiderol Bearings at five points of maximum radial load in this Warren - Quimby Vertical Screw Pump. The Guiderol Bearings have more load carrying capacity in the limited space available and enable vertical mounting of the pump without roller skewing or locking.

KEARNEY & TRECKER FINDS LIFE FACTOR INCREASED IN TABLE FEED APPLICATION



The life factor of the shaft bearing on the table feed screw of Kearney & Trecker Milling Machines has been increased by the simple process of changing to Guiderol Bearings. These bearings increased capacity, performance and bearing life.

BEARING SELECTION GUIDE

A new 140-page Bearing Selection Guide, complete with 30 pages of vital engineering data, has been released by the McGill Manufacturing Co. Ask for Catalog No. 52.

McGILL® — Precision Bearings

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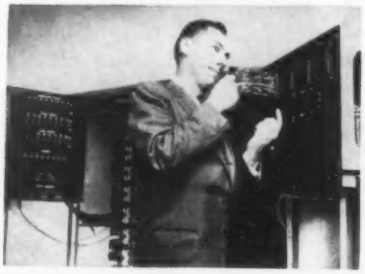


ALWAC's electronic components are arranged into standard plug-in units to minimize maintenance. Using this built-in test box all components in the computing circuitry can be given a static and dynamic check in excess of the normal operating requirements in less than an hour

swivel casters house magnetic drum memory, power supply and logical elements. Power supply and memory cabinets are 28 inches deep, 34 inches wide and 64 inches high. Weights are 520 and 380 pounds respectively. Logical element cabinet is 28 inches deep, 48 inches wide and 64 inches high; weight is 500 pounds. Cost figures out to $34.29 per pound.

Borolite Corp. has been formed by Firth Sterling Inc., American Electro Metal Corp. and Carborundum Co. With headquarters in Niagara Falls, the new corporation will handle combined research and development activities in various metal borides, "Borolites," for high temperature military and civilian applications.

Borides, carbides and aluminides in the Borolite Corp. program show promise of withstanding operating temperatures ranging from 1600 to 2000 F for turbojets and up to 7000 F for rocket nozzles and intermediate applications.

American Electro Metal Corp. has assigned all patents, compositions, processes and techniques to the new corporation and will provide research and pilot plant production. Carborundum Co. will direct production and manufacturing of various borides, since it has experience with abrasive, refractory and powdered materials. Firth Sterling Inc. will be responsible for production know-how and equipment for certain powder metallurgy processes and will fabricate borolite products.

New Process Converts Photo to Line Drawing

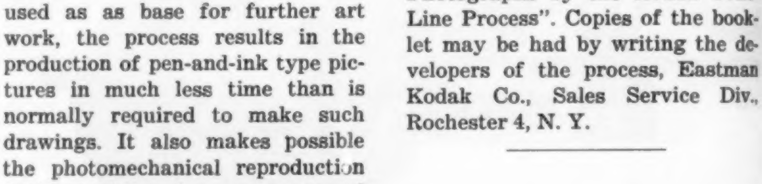
Techniques for converting a continuous tone image into a line drawing by means of a new photographic technique have been developed.

The conversions—which result in an effect similar to that of a pen-and-ink drawing—may be produced from any sharply detailed good photographic film negative. No additional art work is necessary to fit most such pictures for reproduction, but in cases where important characteristics or details must be clearly reproduced these may be enhanced or added by the artists to a tone-line print. When used as as base for further art work, the process results in the production of pen-and-ink type pictures in much less time than is normally required to make such drawings. It also makes possible the photomechanical reproduction of illustrations for many general industrial and other purposes without the use of a halftone screen.

In operation, the Tone-Line process combines a negative with a positive of nearly equal contrast so that the positive in effect acts as a mask for the negative. The negative and the positive are then taped together in register and placed in contact with a sheet of Kodalith Film. The film is then exposed either by rapidly spinning the printing frame underneath a fixed light, or by rotating a moveable light above a stationary frame. Either method allows some light to work its way around the edges of the mask and negative, and produce a line image on the film. This method is quite distinct from solarization methods of producing outlines or from the pseudo-relief effect obtained by using a negative and a positive slightly out of register with each other.

The process is fully described in a booklet titled "Line Effects from Photographs by the Kodak Tone-Line Process". Copies of the booklet may be had by writing the developers of the process, Eastman Kodak Co., Sales Service Div., Rochester 4, N. Y.

Annual Doehler Award for contributions to the advancement of



SIMPLE SOLUTION to the rather knotty problem of transmitting power from the power takeoff of a tractor engine to a unit mounted on a trailer is supplied by flexible shafting. Consisting of a $1\frac{1}{4}$-inch diameter flexible shaft terminated at one end by a telescoping square end tube, the drive is made by Stow Mfg. Co. In addition to absorbing shock loads which would tend to damage the driven mechanism, the telescoping bar and tube serve as a disconnect coupling. A universal joint between shaft and driven device prevents severe bending of the shaft

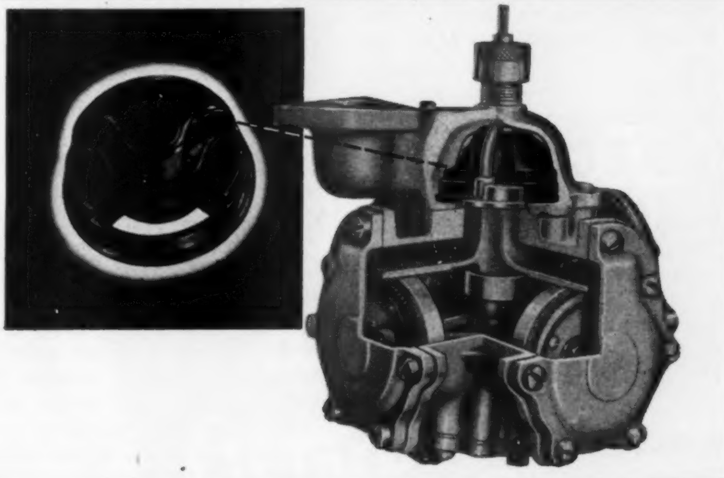
(Advertisement)

WHAT EVERY ENGINEER SHOULD KNOW ABOUT CARBON

Carbon Products Offer Distinct Advantages

Design engineers have profited greatly from the constant development of carbon in industrial applications. For more than 30 years Morganite has continued to explore the possibilities of carbon. Morganite's findings have revealed properties inherent in carbon which, when applied to industrial machinery, simplify design, reduce maintenance costs, and add life to vital equipment.

Today, bearings, valves, valve seats, vanes, gland rings, piston rings, slides and seal noses, all made of carbon, are commonplace. Tomorrow will bring new ideas, extending the use of carbon in industrial design.



Morganite developed and tested a self-lubricating rotary valve suitable for gasoline metering. Tests indicated a 90% reduction in operating troubles formerly due to friction, gumming or sticking.

Carbon Has Unique Features

Unlike metals, carbon needs no oil. The graphitic content of carbon component parts supplies all the lubrication required to operate smoothly and efficiently, even when submerged in water, acids, or other liquids. Carbon will not corrode. It is resistant to most chemicals, alkalis, solvents, gasoline and water.

Morganite carbon will withstand high temperatures and thermal shock. It will retain its strength and hardness under almost any condition.

Morganite carbon is particularly well adapted to the food and chemical industries where freedom from contamination is vitally important. Carbon will not give off unwanted odors or tastes.

Carbon Can Be Accurately Machined

Morganite carbon can be machined to accurate limits and lapped flat to 3 light bands. It can be plated, cemented, or can be permanently vulcanized to natural or synthetic rubber. It can be coated with porcelain or other ceramic material.

Most applications possess distinctive characteristics. To provide exceptional performance for each particular type of service, several grades of Morganite have been developed. When considering the use of a carbon part, it is important to take into account speed and load-radial or axial, the material to be handled, pressures and temperatures, as well as the type of material to which the carbon will be mated. A discussion of these details with a Morganite service engineer will be helpful.

Literature On Request

To help design engineers in solving their many and varied problems, Morganite will be glad to send, without obligation of course, its DESIGN DATA REFERENCE. This brochure contains hints in designing with carbon, charts and other usable data needed in considering carbon applications.

Morganite technicians have many years of experience in developing carbon applications. Their knowledge will be found invaluable. If you desire, they will be glad to discuss your design problems with you.

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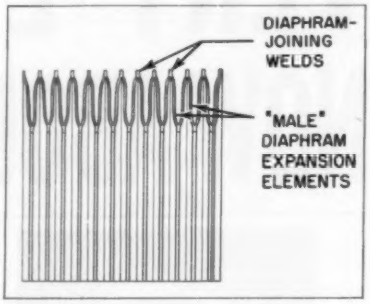
City__________ State__________

the die casting industry and process will be in September at the Annual Meeting of the American Die Casting Institute. The Award is made for the outstanding contribution to the advancement of the die casting industry, or to the art of die casting by technical achievement of a metallurgical or engineering nature, advancements in plant operations, or other activities resulting in the enhancement of the reputation and acceptabilities of die casting.

The Award consists of a suitable plaque and an honorarium of $500 or more, at the discretion of the Award Committee. Nominations for the Award and supporting papers or other material must be received by the Award Committee, American Die Casting Institute, 366 Madison Ave., New York 17, N. Y., by April 15, 1954.

Bellows Life Increased By Welded Diaphragm Design

Analysis of failure of metal bellows has shown that failure occurs mainly at the bend in each convolution, due to the concentration of flexing at this point during expansion and contraction of the bellows. This flexing concentration leads to work hardening and consequent failure of the metal at this point. New welded diaphragm construction, distributes flexing over the entire height of the convolution thus avoiding the concentration of flexing and the resultant work hardening that leads to bellows failure, according to developers of the bellows, Titeflex, Inc. This construction is also said to increase flexibility by increasing the height of bellows convolutions. The increase of flexibility with height of convolution permits much greater expansion range for every inch of bellows length. Thus relatively long expansion ranges can be achieved in bellows of comparatively short initial length.



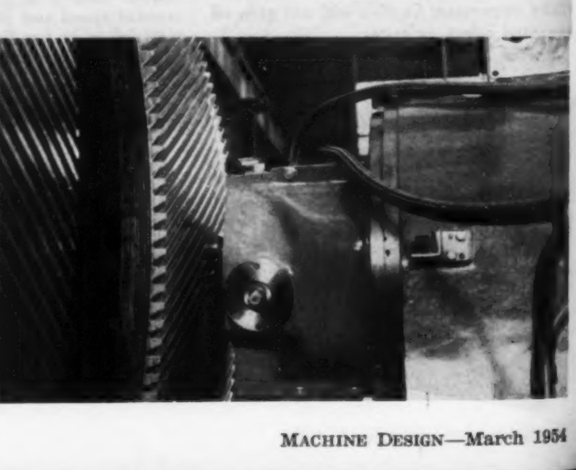
Two types of bellows are manufactured using this new construction. Standard type consists of a series of male expansion elements. The nesting type is a series of nesting male-and-female expansion elements for extreme flexibility applications.

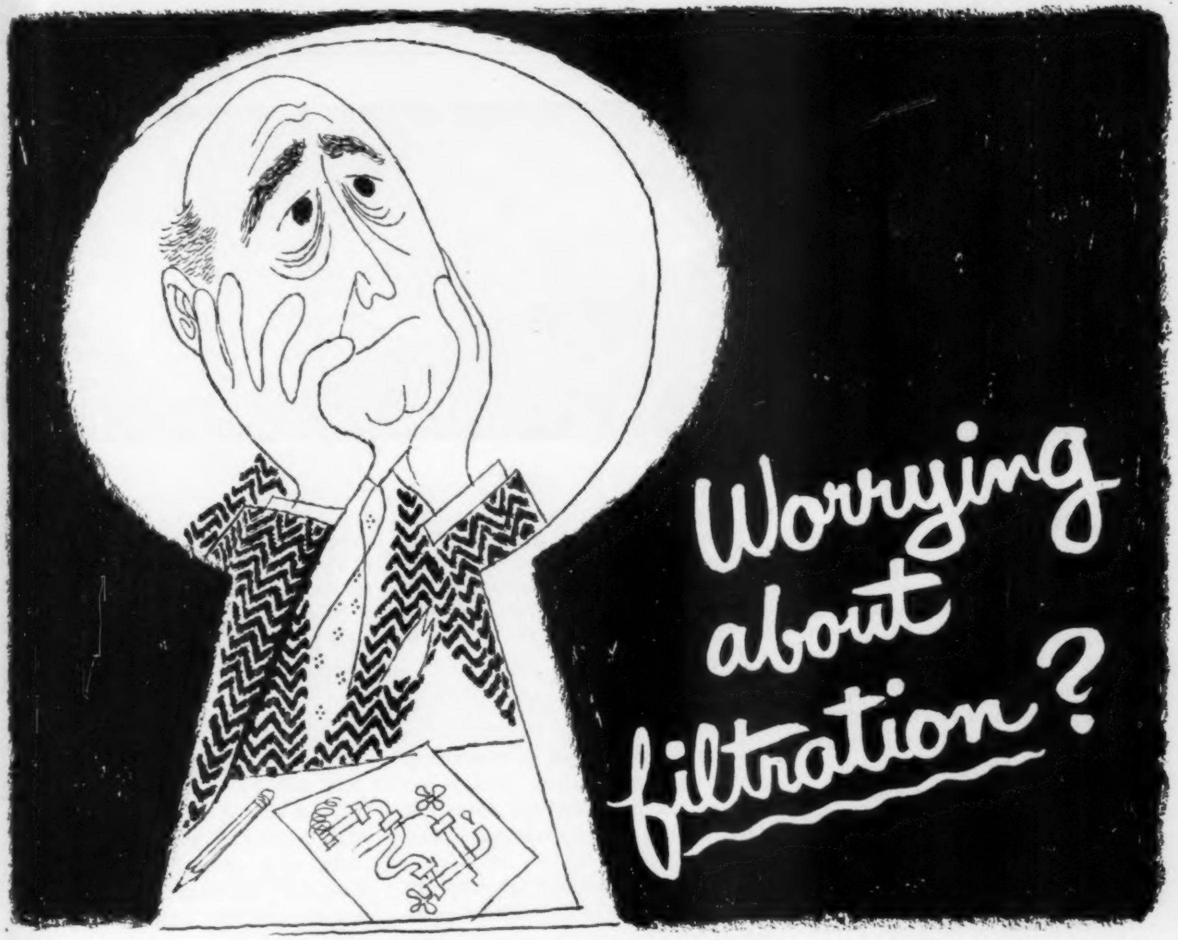
Bellows are manufactured in the male-expansion-element type, in a range of 50 standard sizes from 1.0 inch ID up to 36 inch ID or in special sizes to the customer's specifications. The nesting type can be produced in a range of 27 sizes, from 0.687 inch to 7.375 inch ID. Both the standard and nesting type bellows may be obtained in a variety of metals, such as stainless steel, Monel, Inconel, to suit the particular application or operating conditions. On special order they can be supplied in other white metals with extra high convolutions for additional flexibility, or with any desired wall thickness for additional strength.

Winners in the 1953 International $2000 Prize Competition devoted to "Technical and Research Aspects, Advances and Advantages of the Uses of Lower Melting (lower than parent) Filler Metals in the Non-Fusion Welding Process," have been announced by Eutectic Welding Alloys Corp.

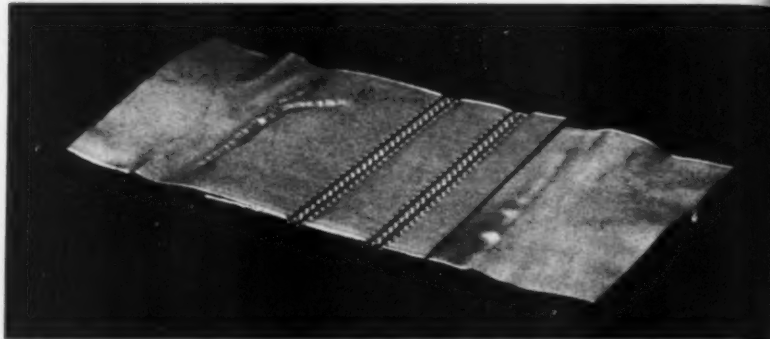
Divided into two categories the

LARGEST TURBINE GEAR SHAVER weighs 130 tons and can shave 220 inch diameter gears, weighing 100 tons. Designed for "crowd" shaving, a superfinishing process for removing minute undulations produced by hobbing, the machine permits selective shaving to obtain 95 per cent bearing across the gear face. Operating drive is to the gear being shaved rather than the cutter, shown closeup at right. Designed by David Brown Machine Tools Ltd., the 33 by 35 by 16 feet high machine is now in use at the Clydbank shipyard of John Brown and Co. Ltd.




Worrying about filtration?

contest included: "A," technical and theoretical and "B," practical applications. Top three winners in the "A" category were Arthur Stuart Laurenson, Belmont Smelting Refining Works, Inc., Ragnar Tidland, Svenska AB Gasaccumulater, Stockholm, Sweden, and H. W. Sandford, United Aircraft Corp. Winners in the "B" category were John Harrison Morton, Baltimore, George Gordon Musted. Sr., Birmingham, England, and Howard A. Hayden, Detroit.



WELDED ALUMINUM FOIL is possible with a process using no heat, electricity, flux or chemical of any kind. A special hand tool, manufactured by the Utica Drop Forge Tool Co., presses the pieces to be welded between special dies. The Koldwelding process can be used with aluminum and electrical copper having an overall maximum thickness of 0.080-inch. Sheet welded by the process is said to meet the same tests used with conventional methods

Punch Press Has Electronic Brain

An automatic punch press with an electronic brain was described recently as "one more step toward the automatic factory of the future" in a talk and motion picture presented at an Institute of Radio Engineers symposium on automation. Developed by General Electric scientists, the machine and its techniques are being adapted to electronic equipment fabrication for the U. S. Signal Corps.

Directions are fed to the punch press by an electronic digital computer. The computer "reads" a perforated card which has information on size, number and location of the holes to be punched. The punch press automatically positions the material to be perforated and performs its punching operations with accuracy of a few thousandths of an inch. Techniques employed to make the punch press perform automatically may well be applied to a number of other industrial operations, including drilling, riveting, stapling, electrical testing and many others.

The punch press is forerunner of an assembly machine to be used in an automatic assembly system being developed for the Signal Corps which will place from 10 to 50 standard components, such as resistors and subminiature tubes, on printed wiring boards at a rate of 30 per minute. This rate can be increased on any one production line by using additional placement machines. The final system will also provide for preparation and testing of components, for transporting them to the assembly machine, and for soldering and testing the completed subassemblies.

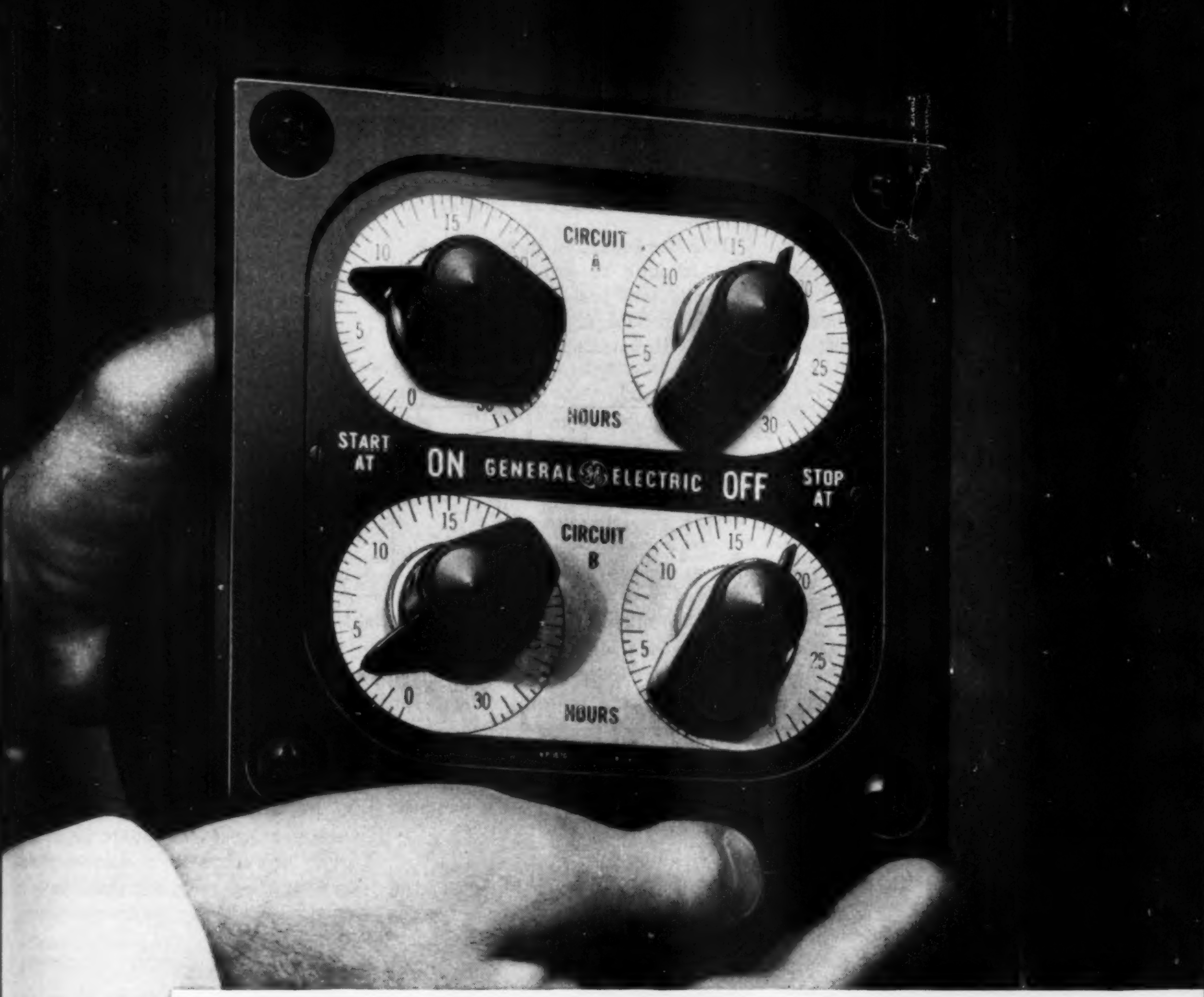
The system is not intended to produce completed products such as radar or television sets, but will produce printed circuit subassemblies for electronic equipment. Subassemblies will be manually combined into complete products, until other machines may be devised to perform the task.

Greatest contribution of the automatic component assembly system will be in improved automation of the small quantity production typical of many military products and specialized commercial lines. While much effort has been applied to the automatic production of electronic equipment to be made by the hundreds of thousands, efforts to increase productivity of job shops have been lacking.

A prime requirement for automation of small quantity production either of electronic assemblies or metal working operations is flexibility. The automatic component assembly system GE is developing for the Signal Corps is being designed for the greatest possible flexibility. A change can be made in the subassembly the system is producing merely by punching new directions on a new program card. No retooling or operator training will be required.

"We've simplified the controls you'll note."

Misco Fabricators, Detroit, Mich., announces the appointment of Garth S. Thompson as President-General Manager. The company has recently been organized as an independent corporation after many years as a division of Michigan Steel Casting Co.



NEW GENERAL ELECTRIC PROCESS TIMER

Offers Greater Flexibility of Timing Control

The new General Electric Type TSA-18 timer simplifies the design of automatic controls for industrial equipment by reducing the need for expensive custom-built timing equipment.

THREE TYPES OF CONTROL—The TSA-18 can be connected for manual start and reset, automatic reset, or automatic reset and repeat operations. Contacts can be set for normally-open or normally-closed operation.

BOTH "ON" AND "OFF" TIME ADJUSTABLE—The new timer provides adjustment for delayed closing, as well as opening, of the main contacts. Thus, a process can begin at any specified time after the timer is energized.

WIDE RANGE OF TIME CYCLES—Ten time cycles are available to give a full dial scale as low as 60 seconds or as long as 30 hours. The TSA-18 can thereby time almost any process regardless of the time length involved.

APPLICATION VERSATILITY—Whatever the process you must control, this new G-E timer will help you meet the timing requirements, accurately and dependably—on baking equipment, plastic and rubber molding, machine tool equipment, heat treating, hydraulic presses, mixing equipment, conveyor loading, chemical processing, photograph processing equipment and many other types of machinery.

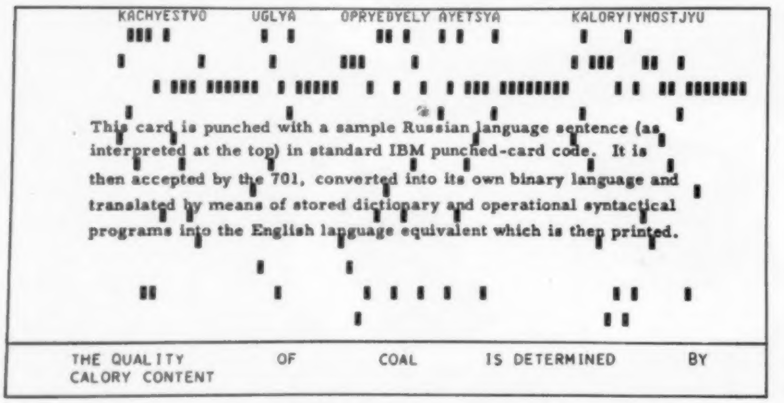
FOR MORE INFORMATION—Call your nearest G-E Apparatus Sales Office; and write for Bulletin GEC-1223, to Section 603-178, General Electric Company, Schenectady 5, New York.

Progress is our most important product

GENERAL ELECTRIC

Big Brain Reads Russian

Russian was recently translated into English by an electronic brain for the first time. Brief statements about politics, law, mathematics, chemistry, metallurgy, communica-



Specimen punched card with the English translation of the Russian sentence, top, in the ruled strip at the bottom

tions and military affairs were submitted in Russian by linguists of the Georgetown University Institute of Languages and Linguistics to the 701 computer of the International Business Machines Corp. Within a few seconds, the computer turned the sentences into easily readable English.

A girl who didn't understand a word of the language of the Soviets punched out the Russian messages on IBM cards. The "brain" dashed off its English translations on an automatic printer at a speed of two and a half lines per second.

More than 60 Russian sentences were given to the "brain" altogether. All were translated smoothly in a demonstration performed jointly by Georgetown and IBM as a phase of IBM's endowed research in computation.

"Potential value of this experiment for the national interest in defense or in peace is readily seen," Prof. Leon Dostert, Georgetown language scholar who originated the practical approach to the idea of electronic translation, declared to a group of scientists and United States government officials who witnessed the demonstration at IBM Headquarters in New York.

Although he emphasized that it is not yet possible "to insert a Russian book at one end and come out with an English book at the other,"—present vocabulary of the machine is only 250 words—Doctor Dostert predicted that "five, perhaps three years hence, interlingual meaning conversion by electronic process in important functional areas of several languages may well be an accomplished fact. Another obstacle to inter-cultural communication will then have been removed—another step taken toward greater comprehension," he noted. "For it is through the print of language that man has ever sought to communicate more widely with his contemporaries, more completely with posterity. Multilingualism has, in part, hindered this quest. Electronic language translation is another stride forward in man's effort to reach his neighbors.

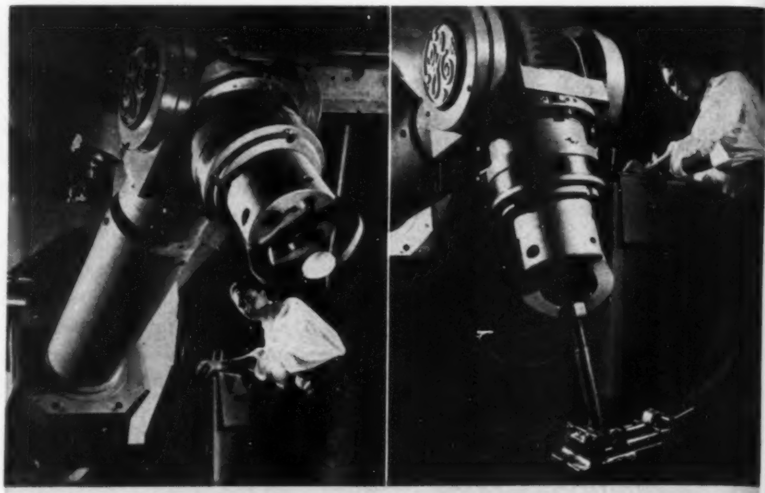
"Concretely, if electronic language translation makes possible, in due course, the translation into the languages of the less developed areas of the world, the basic references and scientific literature in existence in Western languages, this in itself would be significant. The value to research of having current literature in scientific fields readily and promptly available in various idioms is another practical objective."

Insofar as the computer was con-

DRY ICE saves four-inch square rubber pads glued to aircraft surfaces for distributing tension loads in tests. Before the "freeze" method was thought of, the $12 pads were ruined in removal. Now the pads may be pried off in good condition, saving $60,000 per year for Consolidated Vultee Aircraft Corp.

cerned, the demonstration could have been carried out with English and any one of a number of languages. Russian was chosen because present-day understanding of the Soviet by western countries is impeded by the relatively small number of students of Russian as opposed to a steadily growing accumulation of Russian textual material whose true significance cannot even be estimated until its content can be converted into English.

Scientific and technical subjects were chosen for first experiments because that type of writing is done with words having highly specialized meanings, and it is possible to predict that if a word appears in a certain context the chances of its having a certain meaning are extremely high. Consequently, Doctor Dostert assumes that electronic translation will begin with separate dictionaries for each technical area, and that as experience with them grows, enough will be learned to permit accurate translation of our common everyday language, in which are such illogical and unpredictable words as "charleyhorse."



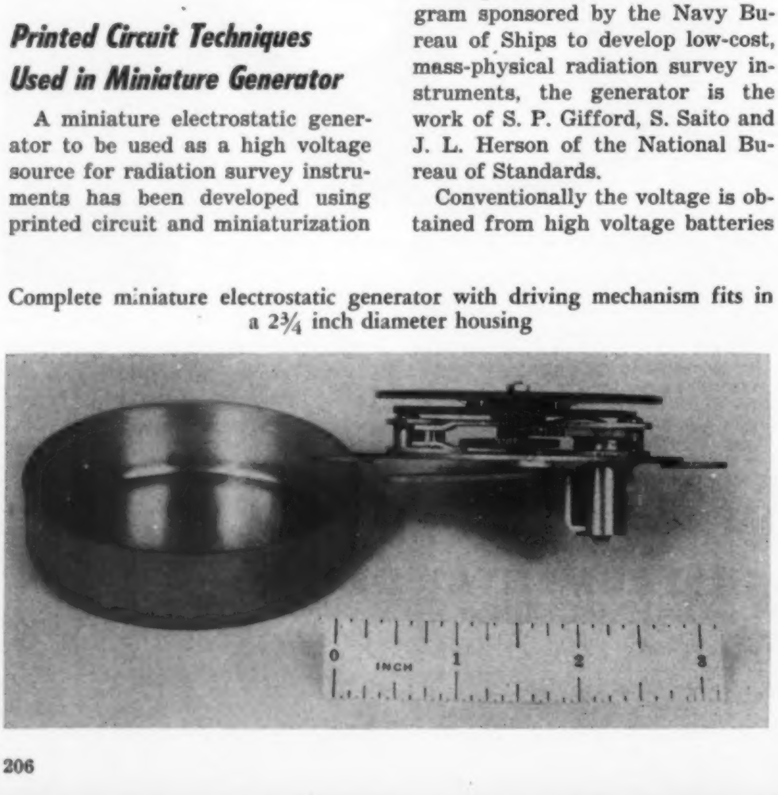
O-MAN is a contraction of overhead manipulator but might conceivably be uttered by someone watching this giant mechanical arm in action. Weighing 15 tons and able to manipulate weights of 7000 pounds, the delicate touch of the fingers permits manipulation of an egg as shown, left. Ability of the "wrist" to rotate through 360 degrees makes operations such as threading a nut or bolt, right, easy. Built by GE under sponsorship of the AEC and the Air Force, the arm will be used to handle radioactive objects. The operator will view his work through special glass and binoculars on telescope from a safe distance when the machine is installed

Printed Circuit Techniques Used in Miniature Generator

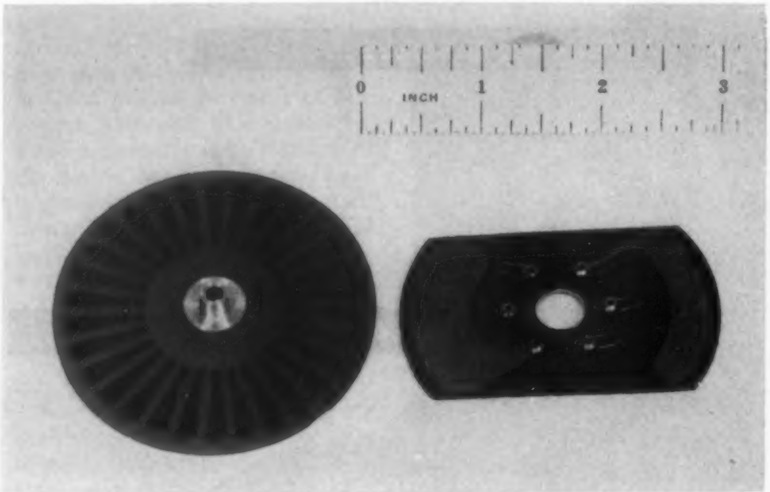
A miniature electrostatic generator to be used as a high voltage source for radiation survey instruments has been developed using printed circuit and miniaturization techniques. One result of a program sponsored by the Navy Bureau of Ships to develop low-cost, mass-physical radiation survey instruments, the generator is the work of S. P. Gifford, S. Saito and J. L. Herson of the National Bureau of Standards.

Conventionally the voltage is obtained from high voltage batteries alone or in combination with capacitors, a vacuum-tube oscillator in combination with a high-ratio transformer and rectifier, or from a vibrator power supply. The new generator designed to replace these is said to be simpler to fabricate, less expensive and avoids the use

Complete miniature electrostatic generator with driving mechanism fits in a 2¾ inch diameter housing



"You mean you only use one side of the paper!?"



Rotor, left, and stator, right, of the miniature electrostatic generator. Parts are made economically by etching copper foil laminated to phenolic material

of difficult-to-obtain batteries. A storage capacitor periodically recharged by the new generator provides the required 2000 volts at 10^{-12} to 10^{-14} amperes.

Basically the generator consists of a stator with two field plate conductors and a rotor with a number of pairs of conducting sectors. Conducting areas are applied to rotor and stator by printed circuit techniques. An attached driving system uses a reciprocating mechanism to drive the rotor at speeds as high as 6000 rpm. Experience has shown that a 0.02-microfarad capacitor can be charged to 2000 volts in about 15 seconds. The capacitor needs recharging only occasionally, which can be done by a single operation of the lever on the driving mechanism.

Largest industrial coolant tank in the world is used to reclarify quench oils at the Ford plant in Canton, Ohio. The 57-foot, 18,000 gallon capacity Magnaflow Separator was built by U. S. Hoffman Machinery Corp.

Designed for continuous operation, fully automatic and self-cleaning, the tank removes 75 to 90 per cent of ferrous solids. In three basic parts, the tank is leakproof, a one piece welded steel unit; the Magnaflow plate containing shielded Alnico permanent magnets, is suspended below oil level where it is said to be 100 per cent effective; the scraper mechanism is a series of scraper flights mounted on a drag chain conveyor, with the motor driven at very slow speed for effective sludge removal.

A new electrical system which makes possible round-the-clock, full capacity high voltage refrigeration for light delivery trucks has been announced by G-E's specialty component motor department.

Five components make up the electrical unit: a heavy-duty alternator which is mounted on a truck engine and is belt-driven from the crankshaft; a small dc motor to drive the evaporator blower; a voltage regulator which maintains constant system voltage completely independent of wide engine-speed variations; a compressor drive motor; and a selenium rectifier to serve double-duty in providing dc voltage to the system from the engine-driven alternator as well as from a 115-volt ac standby source.

GIANT GEAR HOBBER: Spur, helical and herringbone gears as well as worm wheels to 23 feet in diameter can be hobbed on the largest of four standard models of Schiess RF Precision gear hobbers being imported by Kurt Orban Co. Inc. The hobber shown here has a maximum capacity of 18 feet and is shown cutting a 14-foot diameter spur gear

Contact KAYDON of Muskegon

FOR ALL TYPES OF BALL AND ROLLER BEARINGS: 4" BORE TO 120" OUTSIDE DIAMETER

Here's a bearing that can be "custom-engineered" to your exact requirements

KAYDON BI-ANGULAR Roller Bearings are adaptable to various proportions of thrust and radial loads .

IF you've design problems involving various combinations of thrust and radial loads, you don't have to buy separate radial bearings and separate thrust bearings.

KAYDON BI-ANGULAR Roller Bearings normally are built with every other roller reversed, and in such cases, adjacent rollers are at 90° to each other to carry both thrust and radial loads.

To handle thrust loads greater from one direction than the other, the bearing can be made with every second, third or fourth roller reversed, depending on the load inequality.

Whatever *your* bearing needs, KAYDON has all the facilities your engineers require. Whether it's only a few or many special bearings — 4" to 120" outside diameter — or millions of high-precision rollers — contact KAYDON for confidential counsel and cooperation.

31.000" x 34.988" x 2.000" KAYDON BI-ANGULAR Roller Bearings have been produced in large quantities and are successfully in use. They are further proof of KAYDON ability to design and make unusually large precision bearings for specific needs.

Close-up shows alternate rollers reversed. KAYDON BI-ANGULAR Roller Bearings are particularly suitable for low-speed applications involving heavy impact loads.

THE KAYDON ENGINEERING CORP.

MUSKEGON • MICHIGAN

KAYDON Types of Standard and Special Bearings:
Spherical Roller • Taper Roller • Ball Radial • Ball Thrust • Roller Radial • Roller Thrust • Bi-Angular Bearings

PRECISION BALL AND ROLLER BEARINGS

HELPFUL LITERATURE

FOR DESIGN EXECUTIVES

Motor Frame Selection Chart

Westinghouse Electric Corp.—Comparison between old and new NEMA standard dimensions for alternating current motors from 1 to 30 hp is shown on selection wall chart now available. New and old dimensions for polyphase squirrel-cage, polyphase wound-rotor and single-phase foot-mounted motors are presented as well as separate dimensions for flange-mounted motors. Dimensions for drip-proof and totally enclosed, fan-cooled motors are also given.

Push-Pull Controls

American Chain & Cable Co., Automotive & Aircraft Div.—Standard assemblies of Tru-Lay push-pull controls are dimensionally illustrated in folder DH-183, "The Key to Remote Control." Devices transmit positive control impulses directly to any desired point around obstructions, eliminating toggles, bell cranks, devices, rods and other mechanisms.

Three Dimensional Drafting

John R. Cassell Co.—"Three Dimensional Drafting" is title of new presentation that goes deeply into subject of three dimensional or Axonometric drawing. First ten pages are devoted to a technical treatise on the subject, while remainder of catalog is devoted to illustrations of Instrumaster implements used in 3D drawing along with more widely used templates and stencils.

Control Selection Guide

General Electric Co.—Latest selection information on motor starters and pushbuttons is provided by 6-page bulletin GEA-6061. Alternating current magnetic motor starters of 1/6 to 50 hp, ac combination starters of ½ to 25 hp, pushbutton stations, ac manual motor starters and magnetic reversing controllers are covered. In addition to ratings and prices, photos, circuit diagrams and application data are given as well.

Dampening Noise

Natural Rubber Bureau—Various ways in which rubber has helped industry, engineers and architects to offset the stress of noise are pointed up in current issue of "Rubber Developments." Suggestions are offered for deadening noise in buildings, machines, household and in many other situations.

Welding Electrodes

Air Reduction Sales Co.—Over 30 different electrodes—stainless, mild and high tensile steels, cast iron, nonferrous, low hydrogen and hardfacing—are described as to chemical analysis, procedure for welding and application in 50-page pocket guide 1318. Other sections cover picking the right electrode, mechanical properties, testing of electrodes and AWS-ASTM specifications.

Nonferrous Centrifugal Castings

Shenango-Penn Mold Co.—Comparative specifications, chemical analyses and physical properties of nonferrous alloys are tabulated in 4-page bulletin No. 154. Tin, aluminum and manganese bronzes; brass; silicon bronzes; special alloys; and Monel metals are covered.

Identification Signs

Ingram-Richardson Mfg. Co.—Porcelain enamel identification signs for machinery, equipment and trucks are featured in 4-page descriptive bulletin 1153. Sign surfaces are unaffected by dirt, grease, heat and other conditions and remain new-looking with little maintenance.

Electric Immersion Heaters

Cleveland Process Co.—Described in 4-page folder on electric immersion heaters is the heavy duty Clepco-Glorod steel-sheathed unit for heating noncorrosive liquids and mixtures and the acid tank heater which features noncorrosive fused quartz body. Former unit has nickel alloy resistance element covered by quartz body. Accompanying data simplify estimating heat losses, determining kilowatts needed and selecting proper heater size for tank capacity.

10. V-Belt & Sheave Drives

Pyott Foundry & Machine Co.—The necessary selection tables for V-belts and sheaves as well as dimensional illustrations of various types of power transmission equipment are contained in 56-page catalog V-1000 on Vee-Tex drives. All information necessary for specifying Pyott drives is provided in this reference book.

11. Relief Valves

A. W. Cash Valve Mfg. Corp.—"Your Handbook on Temperature and Pressure Relief Valves" is pocket-size pamphlet which explains and illustrates the function of relief valves, both temperature and pressure, insofar as industrial and domestic water heaters are concerned. It touches on both AGA and ASME code requirements, and lists 25 "do's" and "don'ts" for relief valve users.

12. Machine Tool Automation

Turchan Follower Machine Co. — 24-page brochure on automation of machine tools covers hydraulic duplicating attachments for standard machine tools; designing and building complete control systems for machines; and complete new tracer-controlled machine tools. Advantages and applications of servo-control systems are depicted.

13. Electroforming

Bone Engineering Corp. — "Electroforming for Solving Complex Metal Forming Jobs" is title of 8-page brochure which defines electroforming, tells where it is used and where else it can be advantageous. Company's facilities for electroforming are pointed up.

14. Tools & Tool Components

Reid Tool Supply Co.—46-page pocket size catalog lists and illustrates line of tools and tool components. Products include control handle balls, blow guns, chucks, clamps, die components, fixture locks, handles and knobs, hand wheels, milling attachments, safety pliers or tongs, setup aids, tool components, tool and die maker's aids and vises.

USE THE POSTAGE FREE CARDS BELOW for copies of anything listed on this or following helpful literature pages—or for further information on any items in the new parts section or the engineering department equipment section which you'll find on the next several pages.

For additional information on anything advertised in this issue, use the yellow cards in the front section of this magazine.

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15. Solenoid Valves

Eclipse Fuel Engineering Co.—Specifications and prices of DO solenoid valves are highlighted in data sheet M-500-T. Used where control of gas, air, water or oil is required, valves are available in both brass and aluminum for operation up to 150 psi in pipe sizes up to ¾-in.

16. Plastic Tubing

American Hard Rubber Co.—Physical and chemical properties of Ace-Flex transparent plastic tubing are given in 4-page illustrated bulletin 66-D. Standard sizes, which range from 0.120 to 1-in. ID, are covered, together with suggested applications in general industry, processing and laboratory work.

17. Stainless Corrosion Resistance

Babcock & Wilcox Co., Tubular Products Div.—Information on the comparative corrosion resistance of stainless steel tubing and pipe is contained in 8-page data folder TDC 160. Several hundred corrosive media at various temperatures and concentrations are considered.

18. Contract Plate Fabrication

Consolidated Iron-Steel Mfg. Co., Republic Structural Iron Works Div.—Verson hydraulic press brake having 1000-ton capacity has recently been added to company's contract production facilities for commercial steel fabrication. Unit has bed length of 24 ft, with ram level control to within 0.003-in. over its entire length, and will bend plate up to 2-in. thick.

19. Hydraulic Components

Hydraulic Press Mfg. Co.—Set of nine catalogs includes information on line of hydraulic components intended to simplify planning of hydraulic circuits and ordering of hydraulic components. For each product described there are detailed dimensions, cross sectional views, model numbers, ordering information and many other data. Catalogs classify line as to type, style, product and pressure range. Included are pumps, motors, power units, cylinders, valves, filters, gages, pressure switches, flanges and slip joints.

20. Precision Metal Balls

Coolidge Corp.—Precision chrome alloy and stainless steel balls are subject of 4-page catalog "Coolidge Balls". They are available in a variety of grades to suit most applications and in sizes from 5/32 to 25/32-in. Company specializes in high-precision instrument types.

21. Commutators

Han-Kor, Inc.—Of interest to designers of rotating electrical equipment including motors, generators and dynamotors, 16-page illustrated bulletin is entitled "Modern Commutation for Industry." Bulletin shows personnel and advanced facilities available for solution of your commutator problems.

22. Tungsten Carbide Parts

Sintercast Corp. of America—"Sinterforge Tungsten Carbide Components" is title of 12-page illustrated manual covering a wide variety of custom-made shapes and sizes. Parts illustrated range from ½-gram to 18 lb in weight. Chip removal and impact and wear resistant types are included.

23. Needle Bearings

Torrington Co., Bantam Bearings Div.—Particularly useful to designers and engineers in selection of heavy duty needle bearings in catalog No. 61 are numerous pages of cross-section drawings showing details of typical installations. Data covered include design features, selection, sealing, lubrication and dimensions. Speed and life and hardness factors are also listed, together with capacities.

24. Industrial Wheels

Goddeyne Wheel Products—Available welded or bolted construction in sizes from 6 to 16 in. OD, industrial wheels subject of 8-page illustrated catalog No. They have semipneumatic and solid tires a variety of bearing styles and hub Swivel and rigid-mounted casters are covered.

25. Aluminum Appliance Parts

Reynolds Metals Co.—"Aluminum App Parts" is title of 16-page well-illustrated letin showing range of custom-fabricated ponents typical of contract work done. sibilities of application of this service clude cost reduction, improved products increased marketability. Engineering is described as available.

26. V-Belts

Raybestos-Manhattan, Inc., Manhattan ber Div.—Bulletin 6628 describes Super-V-belts, giving a complete list of sizes belt numbers. Large horsepower capacity resistance to shock loads and length are features of the line.

27. Forgings & Castings

Allegheny Ludlum Steel Corp., Forgings Castings Div.—Smooth hammered forgings composite die sections and cast-to-shape steels are covered in 28-page illustrated let. Shapes available, weight limits, ordering information, chemical analyses heat treating instructions are covered in

28. Rubber Transmission Belting

B. F. Goodrich Co.—Detailed instruc on tools, materials and procedures for ing Plylock rubber transmission belting are included in 36-page illustrated catal In addition, construction features and mended usage of various industrial belts given, with special sections devoted to proof, textile, lumbermill and agricul belts.

29. Valves

Robertshaw-Fulton Controls Co., Bridg Thermostat Div.—Used for electrical or sure control of a variety of fluids incl corrosive and volatile types, Marotta are described in 48-page illustrated cata No. 200. Flow curves and installation agrams are provided.

30. Thermostatic Bimetals

W. M. Chace Co.—36-page illustrated let describes and explains 22 uses of bimet as actuating elements in temperature sive devices. Engineering data for design and selection are included.

31. Metal Hose

Flexonics Corp.—32-page illustrated cata covers corrugated, convoluted and stain steel metal hose and numerous special assemblies, representative of a complete Complete specifications and application are provided for the various types, toget with coupling types and installation infor tion.

32. Vibration Isolator

Barry Corp. — Technical and perform data are given in 4-page bulletin type 915 Barrymount for isolation of tion and structure-borne noise from such speed machinery as motor-generator sets, pressors, grinders, fans and blowers. ratings of these mounting units range 15 to 200 lb per isolator. They efficient isolation at frequencies above

33. Tube Fittings

Parker Appliance Co.—Socket-type lok tube fittings for permanently welded are subject of catalog 4370. They are avail in steel or stainless steel in a standard shape in sizes from ¼ to 2 in. OD. includes unions, adapters, connectors, crosses, gage connectors and elbows. suitable for extreme temperature

(*Continued on Page 214*)

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7	29	51	73	95	117	139
8	30	52	74	96	118	140
9	31	53	75	97	119	141
10	32	54	76	98	120	142
11	33	55	77	99	121	143
12	34	56	78	100	122	144
13	35	57	79	101	123	145
14	36	58	80	102	124	146
15	37	59	81	103	125	147
16	38	60	82	104	126	148
17	39	61	83	105	127	149
18	40	62	84	106	128	150
19	41	63	85	107	129	151
20	42	64	86	108	130	152
21	43	65	87	109	131	153
22	44	66	88	110	132	154

MACHINE DESIGN 3-54
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Cleveland 13, Ohio

Please send literature or detailed information on subjects circled at left to—

NAME ______ TITLE ______

COMPANY ______

MACHINES MANUFACTURED ______

ADDRESS ______

CITY AND STATE ______

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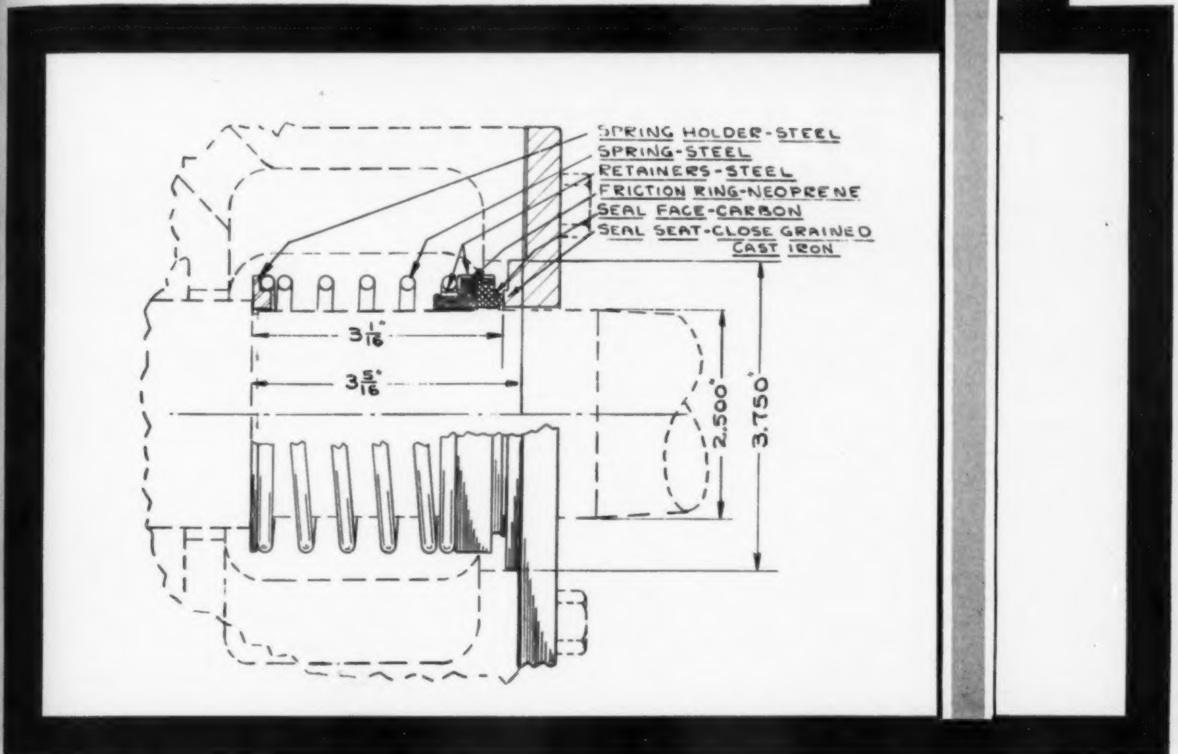
THIS MET A PROBLEM FOR ONE CONDENSING UNIT MAKER—



and ROTARY SEALS can make *Your* seal *sure!*

ROTARY SEALS

Condensing units, confronted with extremely rugged operating conditions in all kinds of refrigerating and air conditioning applications, must stand up to them without breakdown or constant maintenance. It takes a *secure* Shaft Seal to assure that kind of continuous, trouble-free compressor operation. That's why so many leading manufacturers in this as in other fields long ago standardized on ROTARY SEALS —because ROTARY SEALS have proved *on the job* that they mean *Shaft Sealing Certainty.*

Since no two designs pose precisely the same problems, ROTARY SEALS are tailor-made for each specific case, with just the right adaptation of the famous original ROTARY SEAL patented sealing principle to do the most efficient job possible. This principle is clearly explained and illustrated in our booklet, *"Sealing with Certainty"*, a copy of which is yours for the asking.

Manufacturers in many industries assure the long-run user satisfaction which builds more sales by *Sealing with Certainty* with individually designed Rotary Seals. We believe Rotary Seal engineers can save *you* time and money, too—call them in on the first steps of your project!

Sealing with Certainty

THE ROTARY SEAL COMPANY mechanical seals for rotating shafts

2022 NORTH LARRABEE STREET
CHICAGO 14, ILLINOIS, U.S.A.

Helpful Literature

(Continued from Page 212)

34. Magnetism in Cast Stainless

Empire Steel Castings, Inc.—"Magnetism in Cast Stainless Steel" is a 2-page data sheet that clarifies the relative importance of magnetism in chromium and in chrome-nickel stainless steel castings and wrought types. Tables and charts amplify the text.

35. Remote Controls

Teleflex Inc.—Mechanical controls, linkages and mechanisms are subject of 16-page illustrated manual No. 500 which deals with a mechanical system incorporating a rack-like flexible cable within a smooth rigid conduit. Applications and available types of these mechanisms for transferring linear and rotary motion over devious paths are discussed. Engineering worksheets and layout procedures are included.

36. Roller Bearings

Orange Roller Bearing Co.—20-page illustrated engineering data book No. 12 contains complete design, application and performance data on line of Orange Staggered roller bearings which feature exceptionally heavy load carrying capacities. Typical applications are depicted with cross-section drawings.

37. Industrial Products

Johns-Manville—Insulations, refractory products, asbestos cement pipe, packings, gaskets, electrical products, frictional materials, roofing, siding, flooring, partitions and ceilings are some of the products treated in 40-page "Industrial Products" catalog. Essential data for proper application of products are given.

38. Meter-Relay Circuitry

Assembly Products, Inc.—The functioning of eleven control and alarm circuits utilizing meter-relays is explained in 8-page illustrated bulletin 112. Typical of circuits covered are those for battery charging control, null-balance pulsing, voltage or current alarms and temperature control.

39. Automotive Lubricating Equipment

Universal Lubricating Systems, Inc.—36-page illustrated 1952 Automotive catalog deals with complete line of lubricating fittings, drain plugs, pumps and guns, top oilers and accessories for automotive use. Included is interchange chart for standard makes of cars.

40. Aircraft Electric Equipment

Jack & Heintz, Inc.—Among the various types of Rotomotive equipment for aircraft described in 4-page illustrated bulletin 6014 are turbojet and reciprocating engine starters, motor-generators, electric motors and generators, rotary actuators and electrical control systems.

41. Flexible Couplings

Lovejoy Flexible Coupling Co.—20-page illustrated catalog 1000 contains complete design and application data on line of flexible couplings which includes bore sizes ranging from solid to 9½ in., with load ratings from fractional to over 800 hp. Spider type couplings are described with ratings from fractional to 40 hp at 1750 rpm. Cushion type couplings range up from 2.58 hp at 100 rpm.

42. Time Delay Relay

Elastic Stop Nut Corp. of America, AGA Div.—Model NET Agastat time delay relay, designed to introduce either a sequence of time delay periods or a momentary impulse into an electrical circuit, is subject of 4-page illustrated bulletin SR4. It is available in two types: with time delay beginning when coil is energized; or with delay beginning when coil is de-energized. Both are offered with SP-DT or DP-DT contacts.

43. Explosionproof Limit Switches

Minneapolis-Honeywell Regulator Co., Micro Switch Div.—Illustrated 2-page data sheet 84 is descriptive of line of explosionproof switches designed for use as limits, safeties and interlocks in explosive atmospheres. Ratings range up to 20 amp at 460-v ac. Wide range of mountings and actuating mechanisms is offered.

44. Strain Gage

Baldwin-Lima-Hamilton Corp.—Features and specifications of hand type Whittemore strain gage are given in 2-page bulletin 4207. Dial micrometer instrument is furnished for 2 and 10-in. gage lengths. They are compensated for temperature changes.

45. Miniature Connectors

DeJur-Amsco Corp.—Electrical and mechanical ratings, outline dimensions with aluminum hood, schematic illustrations, details on special features and other data are given for series 20 miniature precision connectors with polarizing Screwlock guide pin and socket. They are made with from 7 to 104 contacts.

46. Self-Holding Nuts

Waterbury Pressed Metal Co., P-Nut Div.—Made of hardened and spring tempered steel with turned down corners which bite into metal to lock them in place, P-Nuts are self-holding, self-locking and create tension on bolt or screw. They are described and illustrated in 2-page data sheet. How they can save time and money is detailed.

47. Pilot-Operated Valves

Pantex Mfg. Corp.—Specifications and installation data, engineering drawings and features of Pantex ¾-in. pilot-operated four-way valves are given in 6-page illustrated bulletin 57. Intended for remote control, valves are rated at 100-5000 psi. Six spool types are offered.

48. Alloy Wire, Rod & Strip

Alloy Metal Wire Co.—"A Handbook on Wire, Rod and Strip to Resist Corrosion, Heat and Destructive Service Conditions" is a 38-page illustrated brochure which contains many data on Monel, nickel, Inconel and nickel-clad copper products. General characteristics, special shapes, spring design, pictorial review of applications and other data are included.

49. Applying Electric Heat

Edwin L. Wiegand Co.—Principal industrial electric heating applications and methods are described in 32-page illustrated booklet "101 Ways to Apply Electric Heat." Case histories show tested ways to apply metal-sheathed electric heating units in myriad industrial jobs.

50. Wiring Channel

Stahlin Brothers Fibre Works—16-page illustrated catalog is descriptive of Panel-Chanel wiring system which provides simple, fast means for wiring machine tool and equipment panels. Channel is composed of prepunched laminated plastic side pieces in 2 and 3-in. heights and 3 and 4 ft lengths which can be assembled with 1, 2 or 4-in. width cover plates to form convenient wiring channel.

51. Wire Forms Made Easy

Dudek & Bock Spring Mfg. Co.—Easy-to-understand story on the nature and uses of wire forms is told in this illustrated brochure. It shows how wire forms can do the job of more complicated parts and thus save the user's dollars. Technical language is translated into understandable language. Chart on spring manufacturing variations is included.

52. Wear Strips & Precision Castings

Cadmet Corp.—Two illustrated bulletins deal with precision wear strips made from No. 21 Ampco bronze and with ferrous and nonferrous castings produced to specification by the lost-wax process.

53. British-American Threads

Besley-Welles Corp.—The latest information on British-American Unified Thread standards is contained in illustrated "Tap Reference" manual. Booklet also contains information on tap-drill sizes, selection of taps for various classes of work and types of materials, feeding speeds, and screw thread terms and definitions.

54. Magnetic Amplifiers

Vickers, Inc. Vickers Electric Div.—Modified magnetic amplifiers with ratings up to 11 kw single phase and 45 kw three phase are detailed in three 2-page data sheets. All necessary information for utilizing these units is given. Modifications of standard units can be made to suit requirements.

55. Small Speed Changers

Metron Instrument Co.—Miniature combination variable and fixed ratio speed changers offering variable speed at a reduced nominal output speed in low power applications are subject of 2-page illustrated data sheet No. M1. Variability is 25 to 1, plus choice of hundreds of fixed ratio sections.

56. Pressure Switches & Valves

Barksdale Valves—Manual Shear-Seal valves, solenoid Shear-Seal valves, Meletron pressure switches and Crescent solenoid air valves are subject of descriptive illustrated catalog 3G. All units have their ordering data, flow patterns, characteristics, features and accessories given.

57. Chain Couplings

Browning Mfg. Co.—Line of chain couplings to take the Browning malleable split taper compression bushing in a wide range of sizes with capacities up to and in excess of 150 hp is covered in illustrated data sheet 2017. Coupling specifications and normal service horsepower ratings are given.

58. Mechanical Clutches

Wichita Falls Foundry & Machine Co.—Design features of the Wichita cam and lever type mechanical clutch are expounded in 4-page illustrated bulletin 108. Various models available have maximum starting capacities from 60 to 560 hp. Specification chart is in bulletin.

59. Air Pressure Regulator

Kendall Controls Corp.—Engineering data on improved version of model 10 precision air pressure regulator, known as model 10-A, are offered by this company. Not intended to supplant the present model 10, new model features redesigned shroud or deflector unit and venturi tube to provide optimum balance between internal and external boost. Operating curves are included in data.

60. Small Motors

Dumore Co.—Universal 115-v motors in ratings from 1/20 to 1 hp are subject of illustrated data sheet entitled "For More Power Hours." Specifications of standard and gear motor types are given.

61. Polyphase Horizontal Motors

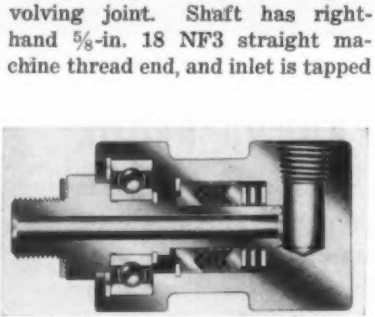
A. O. Smith Corp.—Design and construction features of Smithway polyphase horizontal motors are described and illustrated in this 12-page catalog identified as section 4. Motors feature simplicity of service and mounting, dynamically balanced pressure cast rotor, doubly protected bearings, dual balanced ventilation and cast aluminum terminal box.

NEW PARTS AND MATERIALS

For additional information on these new developments, see Page 211

Revolving Joint

Air or hydraulic piping connections to clutches, power transmission drive units, chucks, spindles, grinding wheels and other rotating machine parts can be made with improved type NV light-running revolving joint. Shaft has right-hand $\frac{5}{8}$-in. 18 NF3 straight machine thread end, and inlet is tapped



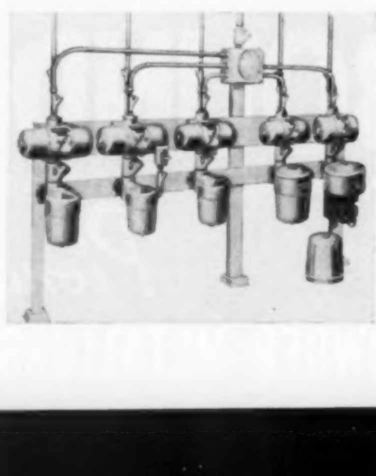
with $\frac{1}{4}$-in. pipe thread. Joint has compact one-piece bronze casing, spring-loaded V-ring seals, hardened and ground steel shaft and permanently lubricated sealed ball bearings. It is suitable for use at temperatures up to 180 F, air pressures to 300 psi and hydraulic pressure to 1500 psi. Recommended maximum speed is 2500 rpm. When installed, joint projects $3\frac{1}{8}$ in. beyond hub. Made by **Barco Mfg. Co.**, Dept. J-33, 500 Hough St., Barrington, Ill.

For more data circle MD-62, Page 211

Enclosures

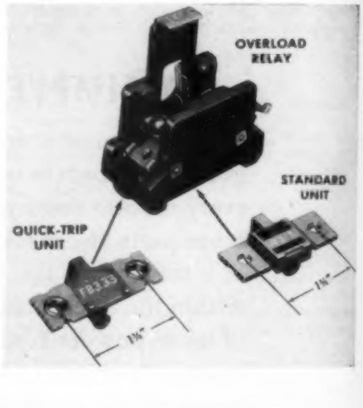
Line of explosionproof, dust and rain-tight circuit breaker, motor starter and line starter combination enclosures, called Unilets, are Underwriters' approved. With an approved sealing Unilet, they may be joined in combination to meet Underwriters' requirements in certain sizes. Each component enclosure for circuit breaker, motor starter and seal has seven-thread engagement at the coupling joints as well as on the bolt-free covers. Safe entrance to individual motor starters for maintenance work is assured in both single and banked combinations in hazardous areas without shutting off other branch circuits. Enclosures are lightweight and have standardized mounting frames for interchangeability of most circuit breakers and motor starters in certain sizes. Straight-through wiring includes versatile hub adaptors which permit use of oversize conduit and conductors. Line covers breaker - starter control equipment rated to 225-amp breaker loads and to 100 hp on three-phase induction motors at 600 v maximum. Made by **Appleton Electric Co.,** 1701-59 Wellington Ave., Chicago 13, Ill.

For more data circle MD-63, Page 211



Overload Thermal Unit

Protection for hermetically sealed compressor motors is provided by type FB quick-trip overload thermal unit. Unit trips in one-third the time required by standard unit. Thermal element

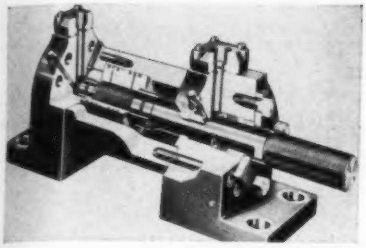


is factory calibrated unit and is not an integral part of the relay block. When starter or control panel is installed, correct thermal unit to match compressor size can quickly be assembled from front. Made by **Square D Co.,** 4041 N. Richards St., Milwaukee 12, Wis.

For more data circle MD-64, Page 211

Hydraulic Cylinders

More than 800 basic models are available in H-P-M line of hydraulic cylinders for service with air, oil and water. Mountings include foot, side lug, centerline, clevis, base, flange, rabbet, trunnion and

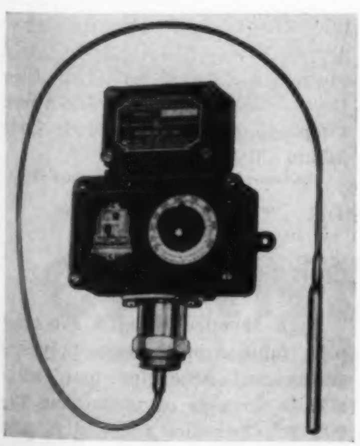


sub-plate styles. Adjustable stroke cylinders are also available with foot or centerline mountings. Heavy-duty units have spring-loaded piston rod packings and are available with male or female, single or double end piston rods, with or without cushion. Standard rods are ground and polished steel, with chrome plated, hardened or hardened and chrome plated rods optionally available. Bore sizes range from $1\frac{1}{2}$ to 12 in. Model 300 series is designed for 150 psi air or 300 psi oil hydraulic service. The 2000 and 3000 series (illustrated) are for 2000 and 3000 psi oil hydraulic service respectively. Made by **Hydraulic Press Mfg. Co.**, Hydraulic Power Div., Lincoln Ave., Mt. Gilead, O.

For more data circle MD-65, Page 211

Temperature Control

Underwriters' approved for use in various Class I and II atmospheres, model VS explosionproof temperature control is of the nonindicating type. It is designed to maintain temperature by controlling the flow of heating agent, and can also be used to control a coolant flow. It can be supplied with a

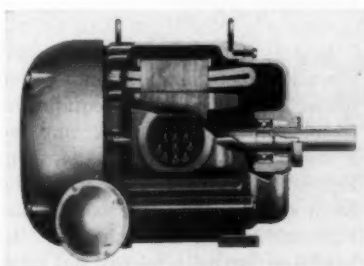


calibrated temperature scale so that control temperature can be set directly on the instrument or with uncalibrated control marked with divisions from zero to 50. Mercury actuated switch is rated at 4 amp at 125 v and at 2 amp at 250 v noninductive ac, for normal service. For pilot duty it is rated at 50 va inductive load. It can be used in combination with a relay and is available with any of ten elements covering temperature ranges from −30 to 1200 F. Control measures $6\frac{1}{2}$ x $6\frac{1}{2}$ x $3\frac{1}{4}$ in. deep and weighs $4\frac{1}{2}$ lb. Made by **Partlow Corp.**, 2 Campion Rd., New Hartford, N. Y.

For more data circle MD-66, Page 211

Magnetic Dustproof Motor

Windings, bearings and other parts of this motor are protected from the harmful effects of fine magnetic dust. A modification of company's CFC Sealedpower totally-enclosed, fan-cooled unit, mo-



tor has self-cleaning action, nonferrous slingers and bushings, heavy fibrous grease seal, prelubricated and sealed ball bearings, and a long, close tolerance fit between bearing brackets and frame. Leads are sealed in with air-hardened plastic. Made by Crocker-Wheeler Div., **Elliott Co.**, Jeannette, Pa.

For more data circle MD-67, Page 211

Check Valve

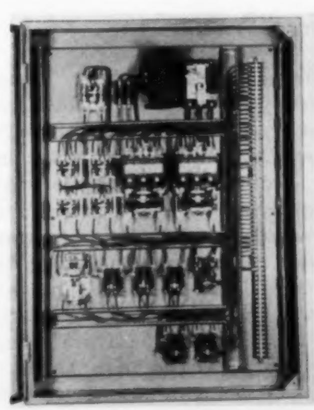
Poppet check valve for high or low pressure pilot operation permits free flow in one direction. Flow in reverse direction is blocked until pilot pressure is applied. Pilot plunger first opens small auxiliary poppet to bleed off pressure, then

main poppet is raised off its seat. Valve has high-strength iron or cast steel body, bronze poppets, O-ring static seals and socket weld pipe flanges. Alternate flanged ports are provided for high pressure inlet when valve is used as a prefill check on large cylinders. Valve is available in sizes from 2 to 8 in. in pressures to 3500 psi. Pilot operating pressures range from 100 to 3500 psi. Made by **Benjamin Lassman & Son,** Route 8, Glenshaw, Pa.

For more data circle MD-68, Page 211

Phenolic Wiring Channels

Panel-Chanels are made of electrical grade phenolic sheet, preformed into perforated channel walls to permit entry and exit of cable wires in panel housings. The moisture-resistant panel material will not warp or sag and can be sawed or cut with tin snips. Lead-off holes $\frac{3}{8}$-in. in diameter are arranged in rows on the panels. Nonperforated covers for channels are fastened to side members with

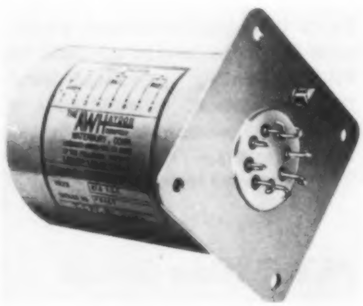


steel bracket clips. Clip slots are spaced every 1 9/16-in. in side panels. T-junctions and end plates are cut from pieces of side section. Panels fit in 1, 2 and 4-in. or other width channels; covers are made in even-inch widths. Illustration shows panels installed without covers. Made by **Stahlin Brothers Fibre Works,** Belding, Mich.

For more data circle MD-69, Page 211

Time Delay Relays

Line of hermetically sealed miniature relays can be supplied with 50, 60 or 400 cycle ac, or governed or standard dc motors. Switch unit

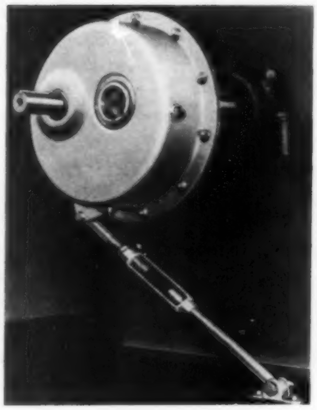


is enclosed in an aluminum housing $2\frac{1}{2}$ in. in diameter and 2 15/16 in. long. Weight is about 1 lb. Three-stud or flange mounting is available, and electrical connection can be made with a glass seal header or AN connector. Made by **A. W. Haydon Co.,** 232 N. Elm St., Waterbury, Conn.

For more data circle MD-70, Page 211

Speed Reducer

Type J speed reducer mounts directly on the shaft to be driven and requires no floor space other than that needed for motor and tie rod connection. It is easily adapted to fit various driven shaft diameters, and offers comparatively high ratios of speed reduction in limited space. Choice of single or double reduction unit makes possible selection of almost any output speed from 14 to 420 rpm by changing sheave sizes. Six

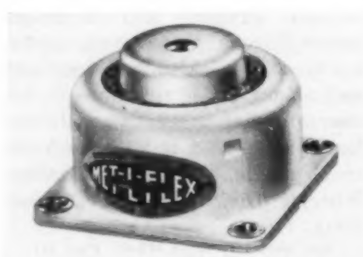


sizes cover range from $\frac{1}{2}$ to 30 hp. Units are of steel construction, with helical gearing and positive lubrication. Made by **Falk Corp.,** 3001 W. Canal St., Milwaukee 8, Wis.

For more data circle MD-71, Page 211

Vibration Mount

Model 1202 cup type metal vibration-absorbing mount for protecting sensitive equipment from vibration and shock employs a resilient cushioning made from knitted wire. The wire is in the form of thousands of continuous vertical springs which are inherently damped and will not pack down or wear out. Top and bot-

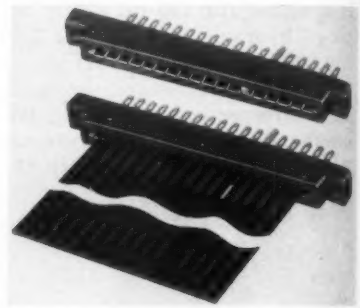


tom-mounted buffer pads provide gradual increase in stiffness and strong damping for positive, negative and radial motion and shock. Unit is corrosion-resistant and operates efficiently over a temperature range from -90 to 175 C. It is available in capacities up to approximately 40 lb and has a $2\frac{3}{8}$-in. square base. Made by **Robinson Aviation, Inc.,** Industrial Div., Teterboro Air Terminal, Teterboro, N. J.

For more data circle MD-72, Page 211

Printed Circuit Connectors

Series PC printed circuit connectors permit direct connection to a printed circuit plug or plug mounted subassembly. Both sides of the printed circuit card can be used for wiring to the external circuit. Connectors designated PC-15, PC-18 and PC-22 accommodate up to 30, 36 or 44 contacts, respectively. Connectors are also available in single row construc-



tion. Multi-conductor two-sided pressure contacts of spring temper phosphor bronze are gold plated over silver for low contact resistance. These low-resistance contacts have a maximum voltage drop of 20 mv at rated currents. Positive polarization is provided with polarizing stud which can be located at any contact. Insulating materials available are mineral filled Melamine, Plaskon reinforced alkyd type 440-A and diallyl phthalate. Available from Electronic Sales Div., **DeJur-Amsco Corp.,** 45-01 Northern Blvd., Long Island City, N. Y.

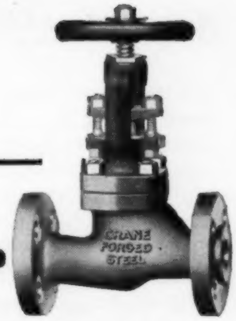
For more data circle MD-73, Page 211

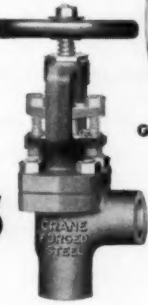
Plastic Pipe

High tensile strength Fibercast pipe, molded of Bakelite polyester resins and Fiberglas mat, withstands stresses of more than 10,000 psi. Corrosion resistant, it also

unmatched in dependability

CRANE

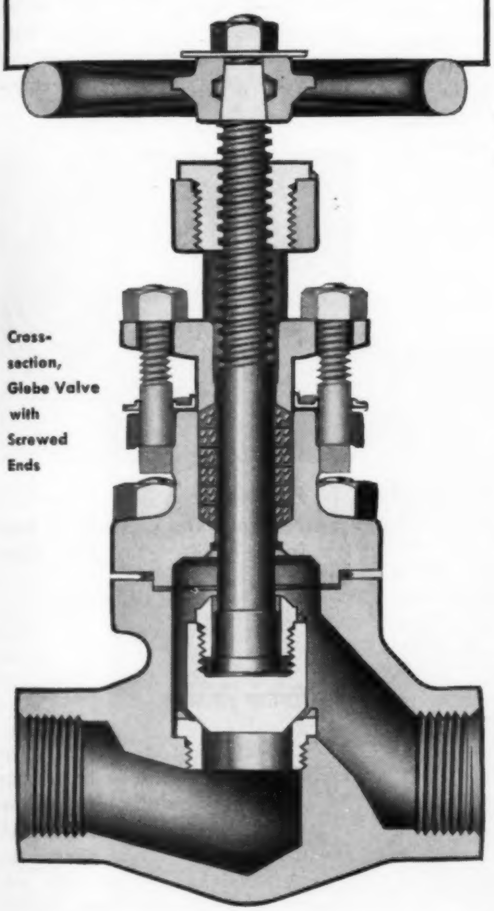




Flanged Ends
Globe and Angle Patterns

Screwed Ends
Globe and Angle Patterns

chlorine valves



Cross-section, Globe Valve with Screwed Ends

now regularly available with screwed ends <u>and</u> flanged ends

Take your choice of patterns in these Crane chlorine valves. They're *Crane Quality* throughout—designed exclusively for water-free chlorine gas or liquid up to 300° F.

In the cross-section you can see their strong, rugged construction—and the narrow bearing 45° taper disc and seat design that provides positive closure. Corrosion-resistant materials are used at all critical points. Disc, body seat ring and disc stem ring are durable Hastelloy "C." The stem and the gasket at the leak-proof bonnet joint are Monel. In the extra deep stuffing box there's laminated packing specially developed for chlorine service.

You're better equipped for chlorine control with Crane chlorine valves. Sizes ½ to 2-inch.

FULL FACTS are in new 4-page folder AD 1976. Write direct or ask your Crane Representative next time he calls.

THE BETTER QUALITY... BIGGER VALUE LINE... IN BRASS, STEEL, IRON

CRANE VALVES

CRANE CO., *General Offices:* 836 S. Michigan Ave., Chicago 5, Illinois
Branches and Wholesalers Serving All Industrial Areas

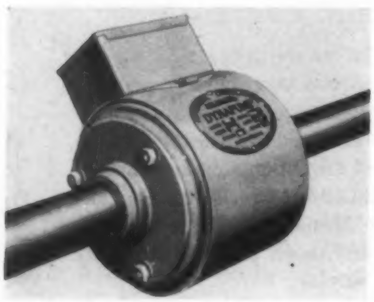
VALVES • FITTINGS • PIPE • PLUMBING • HEATING

exhibits high burst, collapse and beam strengths. Both line pipe and tubing are made in 20-ft joint lengths and outside diameters of $2\frac{7}{8}$, $3\frac{1}{2}$ and $4\frac{1}{2}$ in. A length of the $4\frac{1}{2}$-in. diameter pipe weighs 29 lb. The tubing has an internal operating pressure of 300 psi with ultimate burst pressure of 1200 lb minimum. Line pipe is made for operating pressures of 100 and 200 psi. Recommended operating temperatures range from −65 to 130 F. Available from **Youngstown Sheet and Tube Co.,** Stambaugh Bldg., Youngstown 1, O.

For more data circle MD-74, Page 211

Magnetic Pump

Pump delivers at heads up to 24 ft, is quiet in operation and has no shaft seal or motor bearings to require service. Unit is fabricated

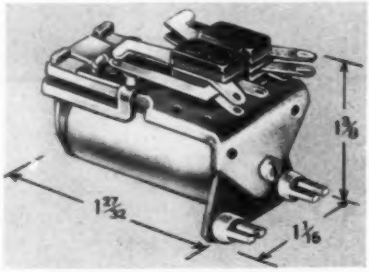


from stainless steel to eliminate corrosion problems and is suitable for either fresh or closed water systems. It accepts interchangeable flanges from $\frac{3}{4}$ to $1\frac{1}{2}$ in. Made by **Fostoria Pressed Steel Corp.,** Fostoria, O.

For more data circle MD-75, Page 211

Precision Miniature Relays

Series of versatile telephone type relays, designated as Class 22 and furnished in variety of hermetically sealed enclosed and open types, is adaptable to low wattage sensitive applications, or to requirements where one relay must perform many switching functions with minimum input power. Units resistant



to shock, vibration and temperature variations can be furnished. Coil and contact spring terminals at mounting end facilitate concealed wiring of either individually or strip mounted relays. Units are available for any voltage up to 440 v ac or up to 230 v dc and can be furnished with single or twin contacts, snap-action contacts, coil resistance from 0.12 to 21,000 ohms, and time delay—slow release to 125 milliseconds. Made by **Magnecraft Electric Co.,** 1442-M W. Van Buren St., Chicago 7, Ill.

For more data circle MD-76, Page 211

Sealed Casters

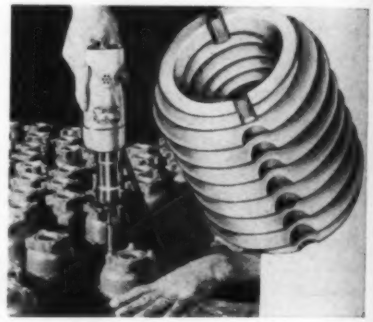
Heavy-duty 3D series H99 casters have main swivel bearing baffle rings which prevent the entrance of dirt and water, grease retainers to prevent loss of swivel lubricant due to vertical drainage, and wheel bearing seals which exclude foreign matter and retain lubricant. Seal design permits flushing out of old lubricant by regreasing with standard pressure lubrication equipment. Casters have fully case hardened raceways, extra heavy king pins

and double ball bearing swivels. Made by the **Bassick Co.,** 437 Howard Ave., Bridgeport 2, Conn.

For more data circle MD-77, Page 211

Threaded Insert

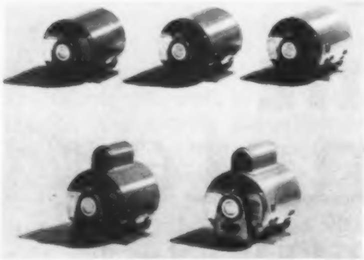
Improved WEG threaded inserts which provide anchor for bolting metal to metal, wood and plastic are now slotted for screwdriver installation in sizes up to $\frac{1}{4}$-in. Larger sizes are installed with insert driver. Inserts are self-tapping in wood, and when cast in molded plastic, their grooves provide locking. They are offered in complete range of sizes, in steel and corrosion-resistant alloy. Made by **Roylyn, Inc.,** 1706 Standard Ave., Glendale 1, Calif.



For more data circle MD-78, Page 211

Jet-Pump Motor

Weighing only 30 lb, this $1\frac{1}{2}$-hp jet-pump motor can be mounted either horizontally or vertically. It conforms to standard NEMA mounting and shaft dimensions and is interchangeable with ratings down to $\frac{1}{4}$-hp. Two NEMA $\frac{5}{8}$-in. diameter shafts are offered, one of cold-rolled steel $1\frac{7}{8}$ in. long with a keyway, the other of stainless steel with a standard thread. Rated at 3450 rpm, the motor is available for 115 or 230 v, 60 cycle, single phase, and 220 or 440 v, 50 or 60 cycle, polyphase. The motor has leads of color coded braidless neoprene, automotive type grease fittings and locked bear-



ings. The insulation system is resistant to moisture, aging, oils, mild acids and alkalies and temperature variations. Made by General Purpose Component Motor Dept., **General Electric Co.,** Schenectady 5, N. Y.

For more data circle MD-79, Page 211

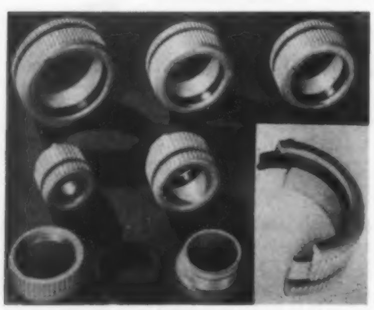
High Capacity V-Belt

Oil-proof, nonspark and heat resistant, Super-Power V-belt has 40 per cent more horsepower capacity than standard V-belts. It has almost no stretch, withstands shock with good elastic resistance, and requires little take-up maintenance and few replacements. Made of synthetic rubber, the belt incorporates a synthetic fiber strength member. Made by **Raybestos-Manhattan Inc.,** Manhattan Rubber Div., Passaic, N. J.

For more data circle MD-80, Page 211

Plastic Pipe Clamp

Molded plastic clamp effects a tight, permanent seal between sections of flexible plastic pipe connected by insert couplings, tees, ells and adapters. Slipped over pipe ends and tightened, the clamp

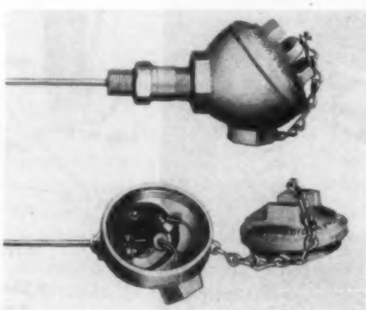


forces ends down onto serrations incorporated in the insert fittings. When clamp is tightened a synthetic gasket is forced against the pipe throughout its full circumference. Clamp is made in sizes for ½, 1, 1½, 2, 3, 4 and 6-in. plastic pipe. It is tightened by a strap wrench and can be reused. Made by **Carlon Products Corp.,** 10225 Meech Ave., Cleveland 5, O.

For more data circle MD-81, Page 211

Temperature Detector

Designed primarily for service on sleeve-bearing motors as an over-temperature detector, model 166KC resistance temperature detector can be used with auxiliary equipment for temperature recording, indicating and monitoring systems. Assembly and head are waterproofed for weather protection. Concentration of the temperature-sensitive element in the tip of the tube permits installation in sleeve bearings or against flush surfaces. Tube is available in

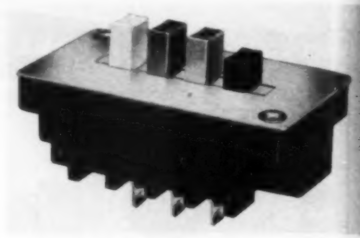


lengths up to 4 ft with a maximum diameter of 0.218-in. Detector element can be withdrawn and replaced by removing holder cap. Usable temperature range is from −70 to 300 C. Made by **Thomas A. Edison, Inc.,** Instrument Div., West Orange, N. J.

For more data circle MD-82, Page 211

Multiple Circuit Switch

Ark-Les model 2930 switch, for pushbutton control, is Underwriters' listed for operation at 1⅛ hp,

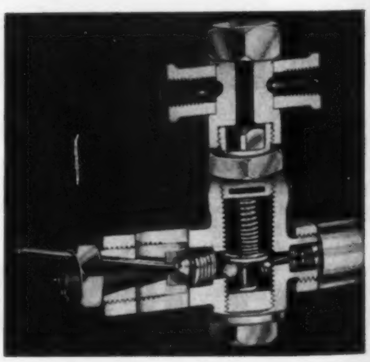


115 or 230 v ac. It is available in 3, 4, 5, 6, 7 and 8-button styles for selection of two to six circuits and off. Switch can be furnished with either binding-screw terminals or quick-connect male terminals for wiring connection. Phenolic pushbuttons are available in a variety of colors. Made by **Ark-Les Switch Corp.,** Dept. 120, 51 Water St., Watertown 72, Mass.

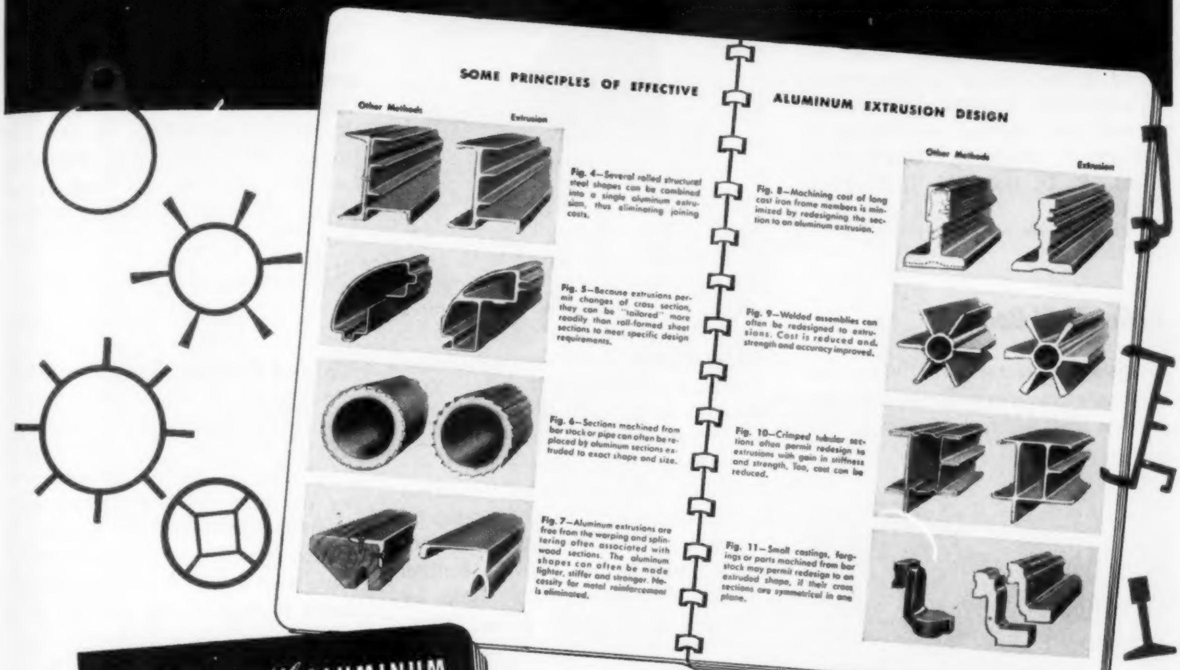
For more data circle MD-83, Page 211

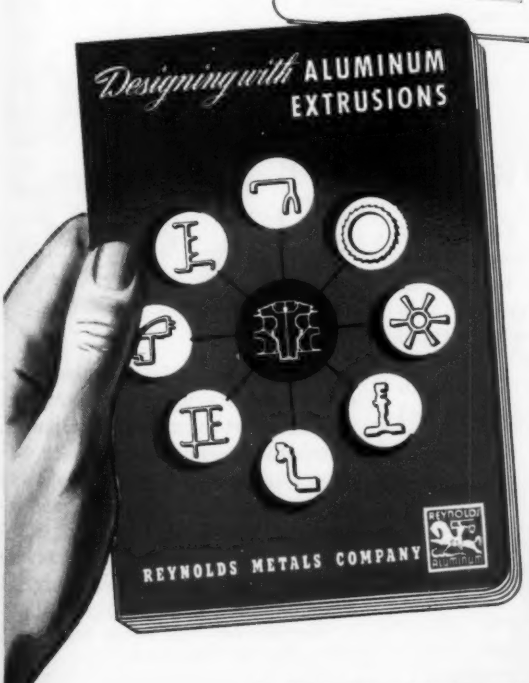
Pneumatic Valve

Wobble-Rod air valve is opened and closed by a stainless steel lever swiveled so that its free end moves vertically, laterally or in arcs and circles. Spring-loaded in the normally closed position, the valve is shut off when the rod is at its top position. Air passage is possible only when the rod is moved to the bottom position. Manual, mechanical or electrical actuation is possible. The valve may be operated by rotating, oscillating, toggle, lever, slotted or straight thrust linkage. A built-in exhaust automatically discharges compressed air trapped in the outlet, as well as in connected equipment, when the valve is shut off. Made of bronze, brass and stain-



How to Take Advantage of Aluminum Extrusions





138 Page Reynolds Handbook Shows How to Increase Design Efficiency ... Lower Production Costs

Limitations of rolled structural shapes, formed sheet sections and machined, cast or welded parts are soon forgotten when you thumb through the authoritative handbook, "Designing With Aluminum Extrusions." The many important advantages obtainable with extrusions are shown in the above pages reproduced from the book.

Extrusion dies are inexpensive, so special shapes that use the metal most efficiently can be produced economically. The handbook shows how . . . with 126 illustrations and chapters on Features of Aluminum Extrusions; Examples of Effective Extrusion Design; Manufacturing Possibilities; Shape Design from Manufacturing Standpoint; Structural Design Considerations; Finishes; Cost Factors and a wealth of other essential data.

Your free copy of "Designing With Aluminum Extrusions" is available, along with a complete index of other helpful Reynolds literature, when requested on a business letterhead. Simply write to Reynolds Metals Company, 2521 South Third Street, Louisville 1, Kentucky.

See "Mister Peepers" Sundays, NBC-TV. Consult local listings for time and station.

REYNOLDS ALUMINUM

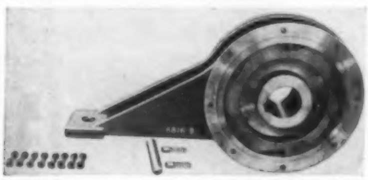
MODERN DESIGN HAS ALUMINUM IN MIND

less steel, the valve is corrosion resistant. A flange located at the center of the valve provides for panel or bracket mounting, or it can be supported on its pipe fittings. Two outlets and two inlets are provided, the latter being located in the ends of a yoke which is adjustable through 360 degrees around the vertical centerline of the valve. Made by **Pantex Mfg. Corp.**, P. O. Box 660, Pawtucket, R. I.

For more data circle MD-84, Page 211

Roller Hold-Back

Enclosed roller type hold-back prevents reversal of bucket elevators and inclined conveyors due

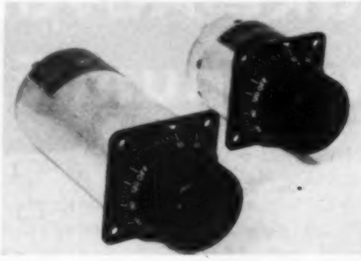


to power failure under load. It consists of an inner member with wedge pockets for six hardened rollers which rotates with the shaft. Rollers wedge against an outer cylinder to prevent reversal of the shaft, releasing instantly when forward motion of the shaft is resumed. The hold-back mounts directely on the driveshaft, with the torque arm bolted to a support with sufficient strength to resist the load exerted when reversal occurs. Maximum torque resistance ranges up to 350,000 lb-in. for units with a maximum bore of 7 in. Made by **Stephens-Adamson Mfg. Co.**, Aurora, Ill.

For more data circle MD-85, Page 211

Intervalometers

A 28-v dc, 3-amp inductive pulse of 0.250-second duration is furnished at regular time intervals by model B-9A and B-10A intervalometers. Time interval between pulses can be changed instantly by turning the dial to interval desired. Re-

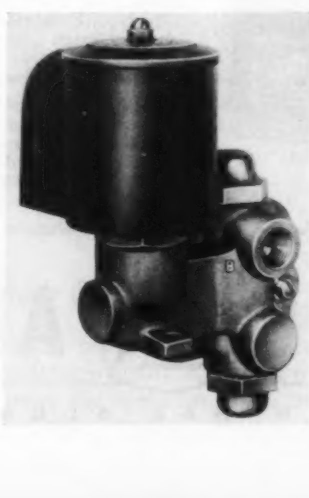


peat accuracy is within ±10 milliseconds over a supply voltage range of 24 to 29 v dc. Each model has two interchangeable time interval scales which provide four ranges to 12, 24, 60 and 120 seconds in 0.1, 0.2, 0.5 and 1-second increments. Auxiliary count limiter, designated as CN-1A1 and shown at right above, can be used as a pulse counter and limiter for stopping the intervalometer at desired number of pulses from 1 to 120 as set on its dial. Two limiters can be connected together to provide control up to 14,400 pulses. Made by **Abrams Instrument Corp.**, Dept. S13, 606 E. Shiawassee St., Lansing 1, Mich.

For more data circle MD-86, Page 211

Three-Way Solenoid Valve

Wide variety of gases and liquids corrosive to bronze, brass and cast iron can be handled at pressures up to 1000 psi by this 8300 ASCO three-way solenoid valve, constructed with either a steel or stainless steel body. Both types have stainless steel trim and

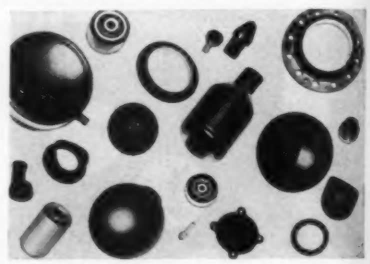


are packless. Valves are employed for applying and exhausting pressure from single or double-acting cylinders. They can be used for direct control to divert and select flow from one or two pipe lines. When equipped with high temperature coils, valves will handle gases to 450 F. Water-tight and explosionproof enclosures are optional. Units have 1-in. pipe size and five port orifices from 1/8 to 3/8-in. Made by **Automatic Switch Co.**, 391 Lakeside Ave., Orange, N. J.

For more data circle MD-87, Page 211

Molded Rubber Parts

Transfer, injection and compression molding are employed to produce custom molded silicone rubber parts to commercial and government specifications. Rubber is bonded to metal to close tolerances, using the newly developed Permadizing process. Parts, offering good



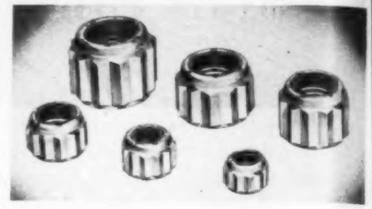
service under adverse operation conditions, are used in automotive, aircraft and electrical equipment. Company's complete service includes compound formulation, mold making facilities and full acceptance of responsibility from blueprint to finished part. Parts are made by **Stillman Rubber Co.**, 5811 Marilyn Ave., Culver City, Calif.

For more data circle MD-88, Page 211

Titanium Locknuts

Line of high-strength titanium alloy nuts meets military tensile strength specifications for steel nuts of the same thread size. Weight of nuts is less than half

New Parts



that of comparable steel nuts. A permanent coating on the nuts provides lubrication and increases resistance to galling and seizure. A nylon insert in the nut collar, smaller in diameter than the bolt thread diameter, provides locking. Bolt entering this collar impresses its threads in the nylon, creating a pressure which holds the nut and bolt threads in intimate metal-to-metal contact. Nuts withstand temperatures up to about 800 F and resist atmospheric and sea water corrosion, as well as the corrosive action of many chemicals. Twelve-point (double hex) nuts are produced in sizes from 5/16 to 5/8-in.; other types will be made available. Made by **Elastic Stop Nut Corp. of America,** 1027 Newark Ave., Union, N. J.

For more data circle MD-89, Page 211

Synchronous Contactor

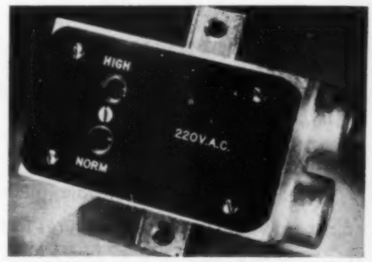
Means of co-ordinating the angular position of a rotating shaft and the opening or closing of an electric circuit is provided by the Rototimer synchronous contactor. It comprises a cam-operated single-pole single-throw switch that can be externally adjusted through 360 degrees in either direction from any starting position. Switch operates

at any point in the rotation of the shaft to which the driveshaft is connected. Electrical system handles 1 amp at 125 v ac. Contactor has sealed-in lubrication and measures 6 3/32 x 5½ x 5 1/16 in. Made by **Farmer Electric Co.,** 21 Mossfield Rd., Waban, Mass.

For more data circle MD-90, Page 211

Voltage Indicator

High-low voltage indicator for either stationary or portable applications is suited for use with equipment whose efficiency is dependent on a specific voltage range. It consists of a Bakelite panel with two glow-lamp pilot lights marked "High" and "Norm" and two test jack receptors mounted in a cast aluminum type FS junction box



with conduit fittings. Model 961 indicates a low of from 95 to 105 v and a high of 130 to 140 v; model 960 indicates a low of 190 to 210 v and a high of 235 to 255 v. Both models work on either alternating or direct current. Units are suitable for operation under conditions of extreme vibration and are also available in other ranges. Made by **Industrial Devices Inc.,** Edgewater, N. J.

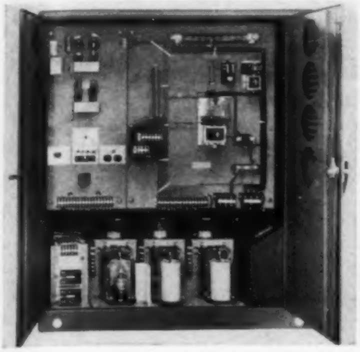
For more data circle MD-91, Page 211

Adjustable Speed Drive

Xatron VS drives are mercury-rectifier, variable-voltage 30, 40 and 50-hp units. Rectification is by a single anode mercury pool rectifier tube which has high life expectancy.

Other characteristics include speed ranges varying between 85

New Parts

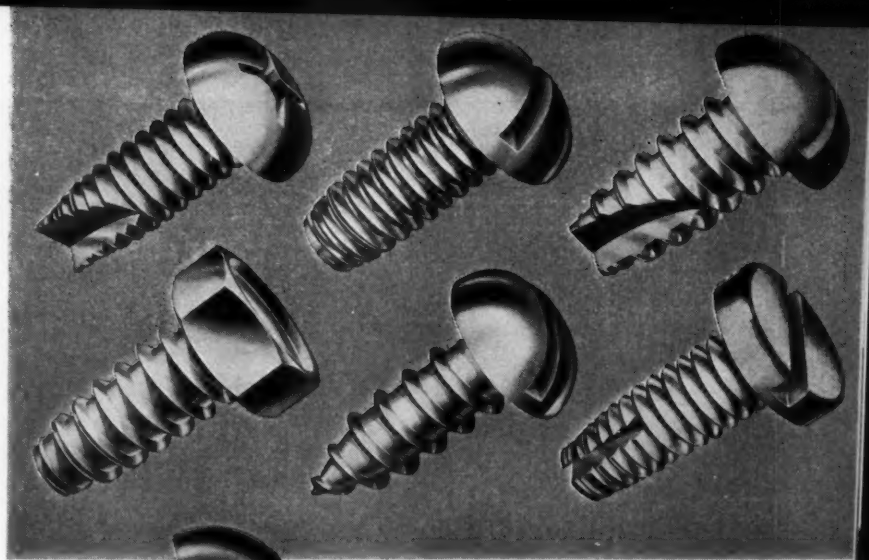


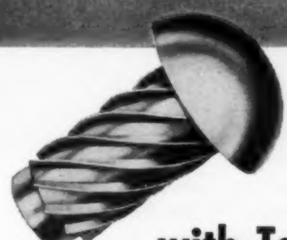
and 2300 rpm, smooth starting, stepless acceleration, close speed control, fast response, high overload capacity, centralized control, quiet operation and low maintenance. Units are also optionally available with dynamic braking circuit; magnetic reversing control; jogging, inching and creeping arrangement and provision for multiple motor drives synchronized as desired. Made by **Reliance Electric & Engineering Co.,** 1088 Ivanhoe Rd., Cleveland 10, O.

For more data circle MD-82, Page 211

Air Pressure Regulator

Series 20AC00 pilot regulator and pressure regulator, an integral unit, reduces line pressures of up to 400 psi to working pressures from 2 to 120 psi. Pilot controlled regulator utilizes regulated air pressure to control delivered air pressure, minimizing the effect of line fluctuations as a factor in deliver-



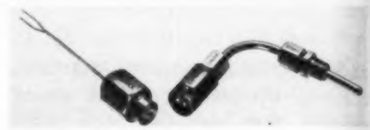


New Parts

ing uniform working air pressure. Unit is made in ½, ¾ and 1-in. sizes. Companion regulator, with same operating features, is available with a remote control unit. Made by **C. A. Norgren Co.**, 3442 S. Elati St., Englewood, Colo.

For more data circle MD-93, Page 211

Thermocouples



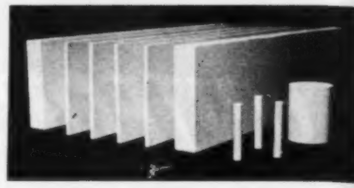
Assembly and disassembly of these quick-disconnect thermocouples is accomplished by plugging the polarized jacks together and turning the nut by hand. Available as a complete unit or as component parts, the thermocouples provide quick-disconnect couplings or complete assemblies. Thermocouples are watertight and vapor-proof. Neoprene seals the lead wires and plug connections; polarized plugs and sockets prevent crossing of thermocouple leads. Plugs and sockets are made of the same material as the thermocouple and will not break the continuity between thermocouple element and lead wires. Small diameter protection tube is made of stainless steel. Open end protection tubes for use with bare wire thermocouples can be supplied for moderate temperatures and pressures. Iron-Constantan, Copper-Constantan and Chromel-Alumel thermocouples are available in 20 gage size. Made by **Conax Corp.**, 4515 Main St., Buffalo 21, N. Y.

For more data circle MD-94, Page 211

Plastic Sheet and Rod

Stycast Hi-K plastic material, possessing dielectric constants of 3, 4, 5, 6, 7 or 8, is available in 1, 2 and 3-in. diameter rods and ½ and 1-in. thick sheets. Maximum variation in dielectric constant is ± 0.15 from 10^6 to 10^{10} cycles, and the dissipation factor is be-

New Parts



low 0.001. Volume resistivity is greater than 10^{14} ohm-cm^3; dielectric strength is greater than 500 v per mil. Physical properties of the plastics vary with dielectric constant, average values being: specific gravity, 1.3; tensile strength, 8000 psi; flexural strength, 13,000 psi; modulus of elasticity, 2×10^5 psi; Izod impact, 0.3 ft lb in. of notch; linear expansion coefficient, 50×10^{-6}/deg C; thermal conductivity, 4×10^{-4} cal/sec/cm^2/deg C/cm; water absorption, 0.1 per cent gain at 25 C for 24 hours; temperature range, -70 to 125 C. Made by **Emerson & Cuming Inc.,** 869 Washington St., Canton, Mass.

For more data circle MD-95, Page 211

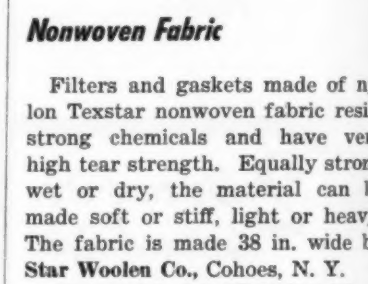
Nonwoven Fabric

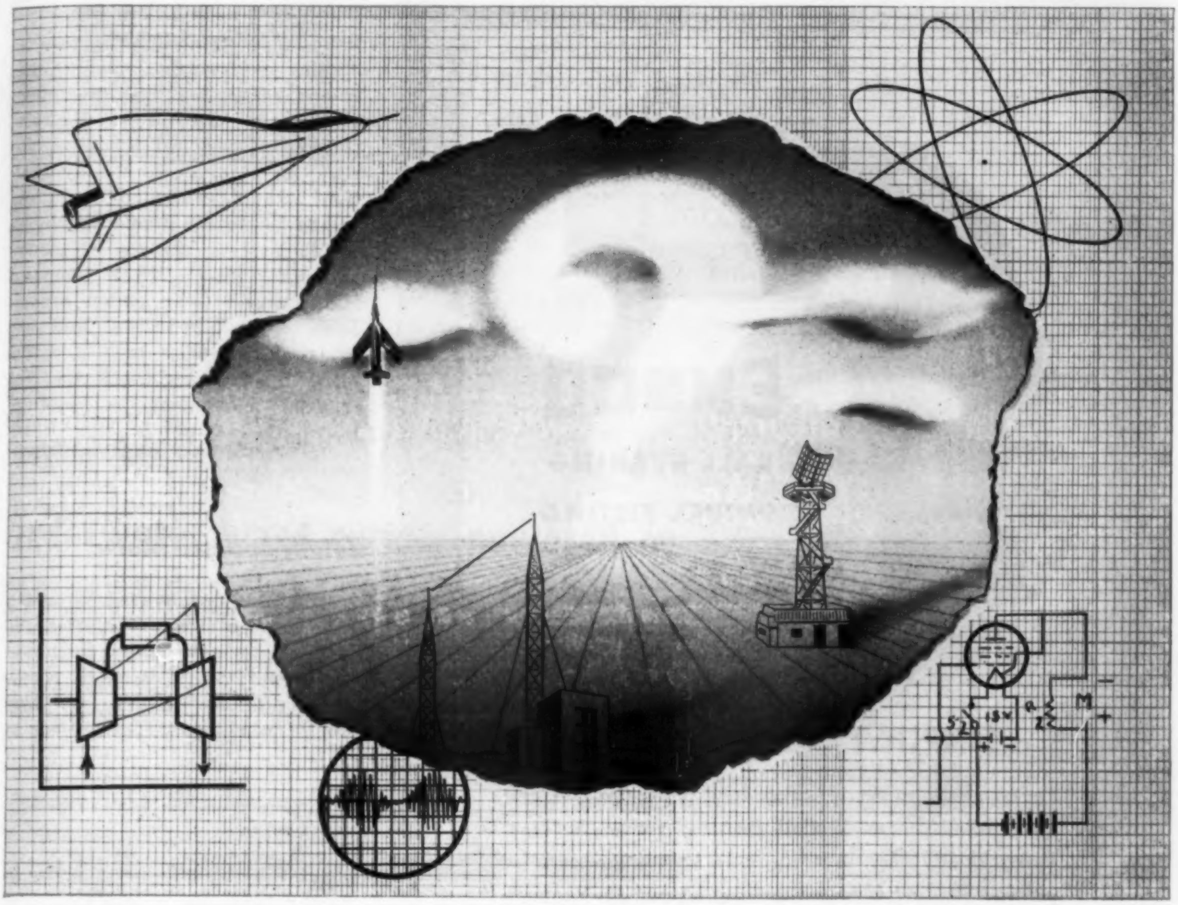
Filters and gaskets made of nylon Texstar nonwoven fabric resist strong chemicals and have very high tear strength. Equally strong wet or dry, the material can be made soft or stiff, light or heavy. The fabric is made 38 in. wide by **Star Woolen Co.,** Cohoes, N. Y.

For more data circle MD-96, Page 211

Miniature Timing Motors

Output speeds for line of miniature timing motors which have less then 2-in. diameter range from 450 rpm to one revolution in 31 days. Motors provide time control for refrigerator defrosters;



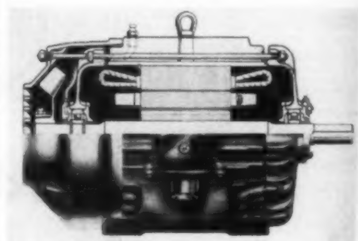


time, temperature and pressure recording mechanisms; washing machines; dryers; time switches; stoker and oil burner controls; thermostats and X-ray timers. Motors can be mounted in any position without effect on operation. Lubricone sealed permanent lubrication system ensures efficiency through −40 to 350 F temperature range. Gear pinion assemblies in each reduction unit are made of fine pitch molded nylon. Made by **Sessions Clock Co.,** Industrial Movement Sales Div., Forestville, Conn.

For more data circle MD-97, Page 211

Totally-Enclosed Motor

Type AK totally enclosed and fan-cooled motor is suitable for outdoor applications and for inside use under moist, dirty or corrosive conditions. It has cast iron single frame construction and is



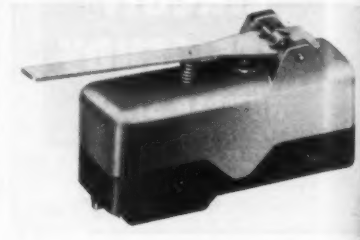
of squirrel-cage induction type. Other characteristics include dust-tight bearings, cast-iron cooling fins and one-piece aluminum fan. Distributed by **Belgian Electric Sales Corp.,** 40 E. 49th St., New York, N. Y.

For more data circle MD-98, Page 211

Snap-Action Switch

Actuating arm of the Unimax type MXT-1 snap-action switch can be furnished in any desired length and with special bends. Usable where low operating force is required, switch has a rigid lever arm pivoted on a bearing pin set in integral supports. Actuator can be modified by eliminating the coil spring to permit operation

New Parts



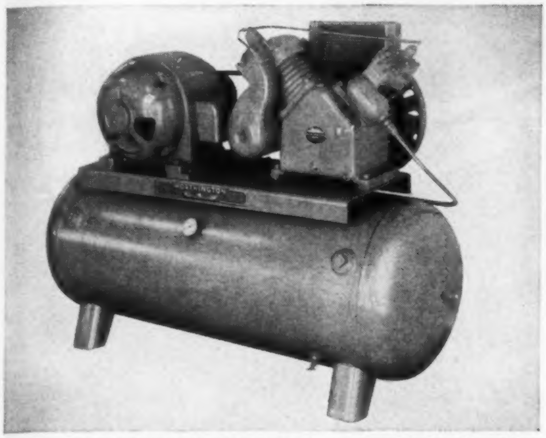
by forces down to 15 grams (model MBT-1). Because of long overtravel provided by lever arm actuator, switch is suitable for cam or slide-operation. Current rating is 15 amp at 125 v ac. Made by **W. L. Maxson Corp.,** Unimax Switch Div., 460 W. 34th St., New York 1, N. Y.

For more data circle MD-99, Page 211

Liquid Level Controls

Sensitivity, accuracy and performance of Electro-Level floatless liquid level controls are unaffected by temperature, pressure, acids or salts. They provide positive control for liquids of low and high conductivity, at any distance. The magnetic type, model LM, utilizes a positive action, heavy-duty, dc-operated relay for the load circuit. A few milliamperes of alternating current through the liquid pilots the device, eliminating electrolysis. Highly sensitive electronic model LE is suited for low conductivity liquids. Positive action is obtained with probe circuit resistance as high as 200 megohms. Both units can be used as high or low level cutoff, alarm or differential con-

COMPRESSED AIR...
do you get it economically?

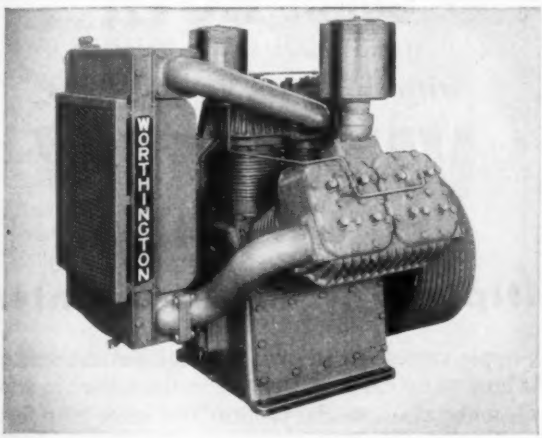


FOR SMALLER QUANTITIES of compressed air, Worthington's Balanced Angle Compressor is your best bet. Mills, machine shops and laundries all over praise the quietness and efficiency of this dependable unit. Sizes range from ⅓ through 15 hp.

If you can't answer this one, you might find some investigation worthwhile.

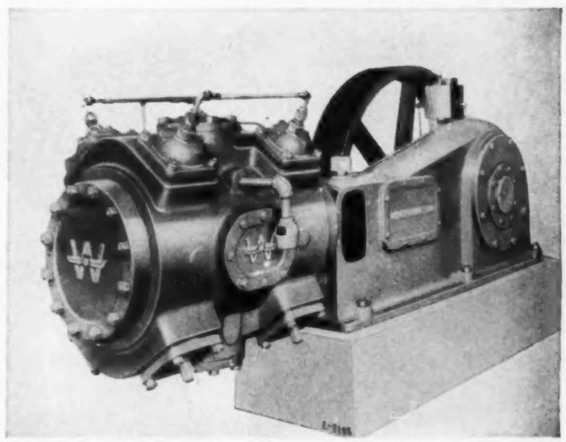
When is a compressor worn out? What model and horsepower is the best answer for your application? How much reserve capacity should you allow for?

Providing the answers to these and other questions about air compression is all in the day's work for Worthington. We've got the widest line to choose from, so you'll know any new Worthington equipment you get is "right" for your particular job.



COMPACTNESS AND EASY INSTALLATION are the big advantages of Worthington Radial Air Compressors, used where larger quantities of air are needed. Like all Worthington compressors, the radial is equipped with our exclusive Feather* Valve. Sizes from 25 to 100 hp.

*Reg. U. S. Pat. Off.



FOR THE REALLY BIG JOBS, specify the horizontal, water-cooled HB—the work horse of the Worthington compressor line. We've had reports about 30-year-old HB's that are giving the same good service as when first installed. Sizes from 7½ on up to 125 hp.

Get the whole story about these modern compressors. Write for Bulletin WP-1099-B52 to Worthington Corporation, Pump and Compressor Merchandising Division, Harrison, New Jersey.

WORTHINGTON

PC.3.14

SPECIFY THESE WORTHINGTON STANDARD PRODUCTS ON YOUR EQUIPMENT

Compressors • Pumps • Multi-V-Drives • Variable Speed Drives

Ham on Rye or Rye on Hand?

Count BOTH

(and 1,001 Other Items)

with this New Veeder-Root

VARY-TALLY

Multiple-Unit Reset Counter

Knowing how many apple versus lemon pies, for instance, are ordered on a given day . . . or how many cases of what are in the cellar . . . helps a restaurant *count*rol, make plans, make profits. The same goes for a manufacturer or wholesaler seeking tighter inventory control . . . or for any of the following:

- **Traffic Engineers**
- **Schools and Colleges**
- **Nurserymen and Florists**
- **State Park and Forest Services**
- **Laboratories**
- **Milk Plants and Ice Cream Processors**
- **Researchers**
- **Industrial Plants**
- **Textile Mills**
- **Inspection and Quality Control**
- **Jobbers, Wholesalers, Distributors**
- **Restaurants and Hotels**
- **Mail and Phone Order Departments**
- **Laundries and Linen Supply Houses**
- **Manufacturers of equipment for:**
  - **Order Control**
  - **Stock Control**
  - **Inventory Control**
  - **Traffic Control**
  - **Sales and Market Analysis**
  - **Laboratory Analysis**
  - **Payroll Preparation**
- **and What Do You Want to Count?**

Arranged compactly on stands in tiers, the Vary-Tally can be supplied in any of 66 combinations, up to 6 banks high and 12 units wide, with a minimum of 2 units wide. Write for news sheet and prices.

"The Name that Counts" **VEEDER-ROOT INC.**

HARTFORD 2, CONNECTICUT

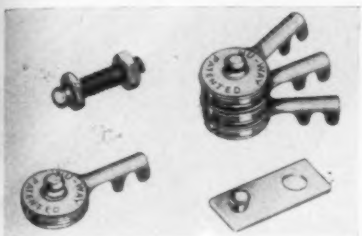
Chicago 6, Ill. • New York 19, N. Y. • Greenville, S. C. • Montreal 2, Canada • Dundee, Scotland • Offices and Agents in Principal Cities

trol. Loads up to 1 hp can be controlled; input is 115 v ac. Made by **Ferrara Inc.,** 8106 W. Nine Mile Rd., Oak Park 37, Mich.

For more data circle MD-100, Page 211

Electrical Terminals

Fabricated from nickel plated brass for weather resistance and rigidity, Nu-Way snap-type electrical terminals are secured by clamping the prongs over stripped wires. Joint can be soldered if desired.



Multiple wires can be joined with good electrical contact by snapping terminals on top of each other. Such connections can be changed, separated or rearranged. Terminals measure 7/16-in. diameter x ⅞-in. long. Made by **John Marshall Associates,** Dept. B, P. O. Box 2463, Bridgeport, Conn.

For more data circle MD-101, Page 211

Molded O-Rings

O-rings molded from Teflon are now available in several standard sizes. They are chemically inert, nonadhesive and can be used over a temperature range from −320 to 500 F. Uniform quality and size tolerances are held. Made by Plastics Div., **Sparta Heat Treat Co.,** East Sparta, O.

For more data circle MD-102, Page 211

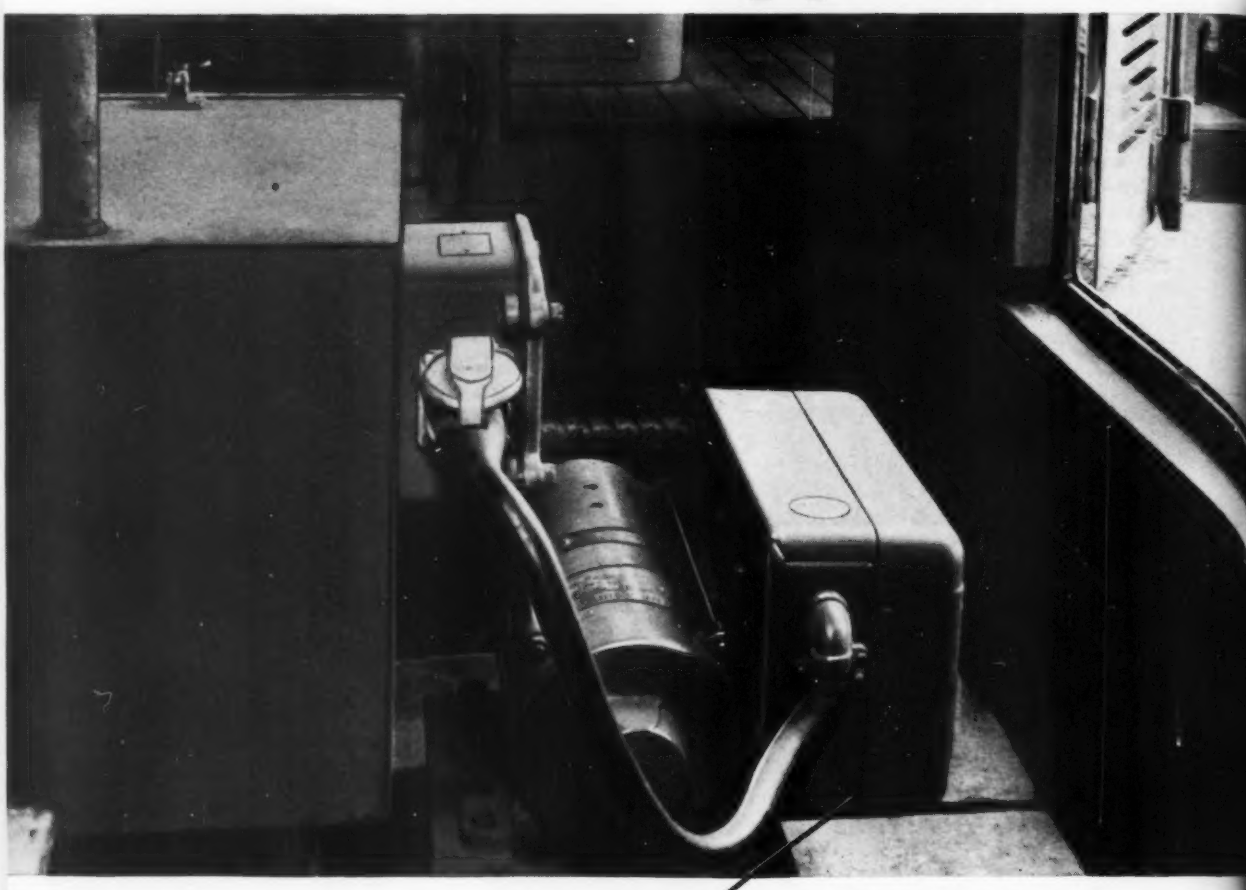
Moisture Separator

Water, oil and entrained solids are effectively removed from horizontal steam, air and gas lines by the 31-N moisture separator. Fab-

(*Continued on Page* 242)

The Erie Railroad story shows why it pays to ...

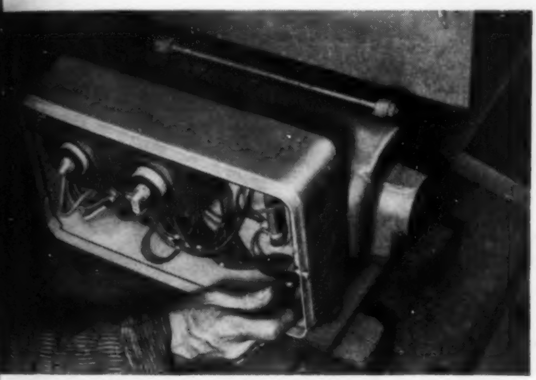
Get motor application



THE ERIE RAILROAD REQUIRED motor alternators to provide 115-volt, 60-cycle power for communication equipment between diesel-electric engines, cabooses, and control stations. The railroad had been using a 64-volt batterv svstem. Illustrations (above) show the G-E designed motor alternator—chosen after competitive tests—and location in the engine. At right, is a brief story of how G-E engineers met the rigid specifications and solved the Erie Railroad's particular problem.

help from G-E Engineers

PROBLEM WAS PRESENTED to G-E sales engineer. Erie Railroad required maintenance-free operation for 3 month periods and a design that would make all servicing feasible except rewinding.



G-E ENGINEERS at the Specialty Component Motor Department in Fort Wayne, Indiana, designed a special open construction type of motor alternator to meet the railroad's precise specifications.

RUGGED TESTS were given this G-E motor alternator on a switcher in railroad's Marion, Ohio, yard. The design struck a happy medium as to design and service, now used on 75 passenger and yard diesels.

PROBLEMS of motor design and motor application are being solved daily at General Electric's Specialty Component Motor Department in Fort Wayne, Indiana. The accumulated engineering experience—unmatched anywhere—starts working for you when you bring your tough motor application problems to General Electric.

THE ERIE RAILROAD had the problem of providing a dependable power supply for their communications equipment mounted on diesel-electric passenger locomotives, switchers and cabooses. They established their needs as a motor-alternator set supplying 115-volt, 60-cycle current. Close voltage control was necessary to insure satisfactory radio tube operation.

WITH THESE REQUIREMENTS in mind, General Electric designed a 2-unit 2-bearing motor alternator which proved the merit and effectiveness of G-E application help in rugged competitive tests.

WHEN YOU HAVE a tough motor application problem, take it to General Electric. Take advantage of G-E engineering skill and talent backed by years of experience. Contact your nearby General Electric Apparatus Sales Office. General Electric Co., Schenectady 5, N. Y. 704-21

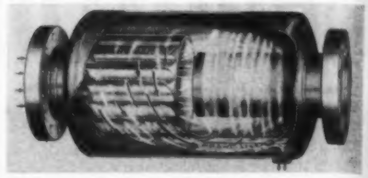
You can put your confidence in—

GENERAL ELECTRIC

APPLICATION HELP provided by G-E engineers in solving the Erie Railroad's problem typifies the many cases where design engineers have found it pays to take motor application problems to G.E.

New Parts

(*Continued from Page 239*)



ricated steel unit is self-cleaning, occupies only slightly more space than section of pipe it replaces and can be used for a wide range of pressures. Made by **Wright-Austin Co.**, 3245 Wight St., Detroit 7, Mich.

For more data circle MD-103, Page 211

Bedways

Various bedway designs, including standard ways and those made to specifications, are made of high grade tool steel laminated to a soft steel backing under pressure.

Ways are hardened to a uniform depth of 3/16-in. and have a Rockwell C hardness of 65-66. Hardened ways and gibs are ground to tolerances of ± 0.0002-in. Made by **Ohio Knife Co.**, Dept. 1, Cincinnati 23, O.

For more data circle MD-104, Page 211

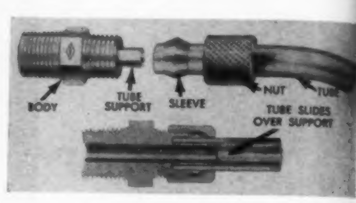
Corrosion Resistant Alloys

Hardenable corrosion resistant alloys are made in cast form. DC-50 castings are produced under a license agreement with Armco Steel Corp., the DC-50 alloy being essentially the same as Armco 17-4 PH hardenable stainless alloy, heretofore produced in wrought form only. In hardened condition, the new alloy has a minimum ten-

sile strength of 180,000 psi and minimum yield strength of 165,-000 psi. It has good resistance to sea water corrosion and pitting and can be used where abrasive conditions exist. Other uses include pumps, valves and thrust bearings. Made by **Donegal Mfg. Co.,** Marietta, Pa.

For more data circle MD-105, Page 211

Plastic Tubing Fittings



Compression type Poly-Flo fittings for connecting polyethylene and other plastic tubing are easily and rapidly installed and can be tightened by fingers only. Fittings consist of a brass body with integral tube support, brass nut and polyethylene sleeve. They can be supplied in stainless steel for use with Teflon or other tubing. Unions, half unions, elbows, tees and bulkhead unions are made. Joints will hold at burst pressures of 0.040 or 0.062-in. polyethylene tubing and are recommended for temperatures from −90 to 175 F for working pressures up to 125 psi in ¼-in. size and 100 psi in ⅜-in. size. Other sizes will be made available. Made by **Imperial Brass Mfg. Co.,** 1200 W. Harrison St., Chicago 7, Ill.

For more data circle MD-106, Page 211

Plastic Coating

Coating of Plastisol 77X-1078 offers protection against corrosive action of trichlorethylene and perchlorethylene. The material produces a hard, slick surface. A primer used under the plastisol provides maximum adhesion and insures against seepage of corrosive chemical under the coating. Made by **Stanley Chemical Co.,** 1 Berlin St., East Berlin, Conn.

For more data circle MD-107, Page 211

A-4284

ater Protection Against Foreign Matter • Long-Life Bearings • Many Tried and Proven Features

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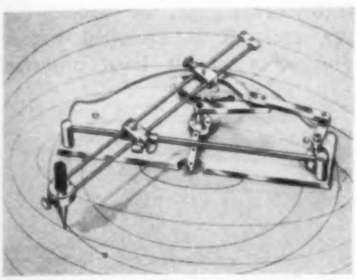
ENGINEERING
DEPARTMENT

EQUIPMENT

Ellipsograph

With a major axis of over 12 in., the Ellipsograph produces mathematically true ellipses. Within the instrument's capacity, all sizes, shapes and projections of ellipses

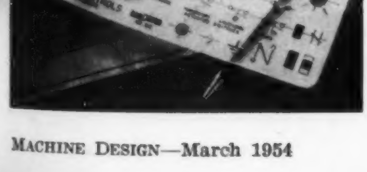


can be drawn with pencil, inking pen or scriber. A linkage system which moves the cross-guide pin on a straight line eliminates all play and friction and provides accurate and easy operation. Made by **Omicron Co.,** P. O. Box 907, Glendale, Calif.

For more data circle MD-108, Page 211

Template

No. 312 electric controls template assists in the depiction of circuits in electrically controlled machinery and electric automation of production processes. Incorporating the March 1953 JIC standard symbols, the template is made of 0.030



matte finish mathematical quality plastic. Printing is on the matte surface and machine-milled cutouts grouped to facilitate drawing combinations of symbols most frequently appearing in sequence. The template measures 9 × 3⅜-in. Made by **Rapidesign Inc.**, P. O. Box 592, Glendale, Calif.

For more data circle MD-109, Page 211

Manometer

For plus, minus and differential readings, I-V gage serves as air filter gage, static pressure indicator, draft gage, air velocity meter, etc. It is equipped with a plunger type oil level adjuster.

Ranges are 0 to 3 in. and 0 to 7 in. water, with low range accuracy of 0.01-in. water. A direct-reading velocity model for use with a pitot tube has ranges of 0 to 7000 and 0 to 10,500 fpm. Housing is molded plastic. Made by **F. W. Dwyer Mfg. Co.**, 317 S. Western Ave., Chicago 12, Ill.

For more data circle MD-110, Page 211

Viewer-Enlarger

Combination microfilm reader and enlarger, the Filmsort Reviewer operates with 16 or 35mm microfilm filed in aperture or jacket cards, or with roll film through the use of an accessory roll film attachment. Three possible magnifications are attainable, depending on the lens used. The

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	Carbon	Manganese	Silicon	Phosphorous	Sulphur
Chemical Comp.	1.00-1.3%	11.00-13.0%	0.30-1.00%	0.07% Max.	0.05% Max.

Tensile Strength	95,000-145,000 psi
Yield Strength (0.2% Set)	45,000-57,000 psi
Elongation in 2 Inches	30-65%
Modulus of Elasticity	28,000,000-29,000,000 psi
Reduction of Area	30-40%
Brinell Hardness Number	170-220 BHN
Work Hardens to	450-550 BHN
V-Notch Impact Strength	
Charpy	90-110 ft.-lbs.
Izod	100 ft.-lbs. approx.

Above properties based on one-inch diameter bars, heat-treated with water quench from above 1832°F.

AMERICAN Brake Shoe COMPANY

AMERICAN MANGANESE STEEL DIVISION
Chicago Heights, Ill.

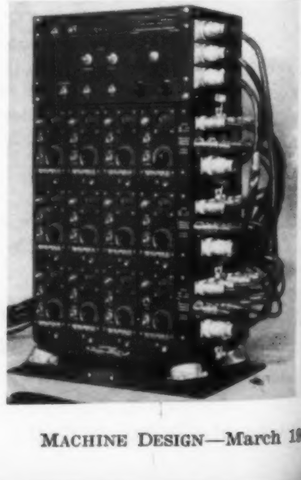
Engineering Equipment

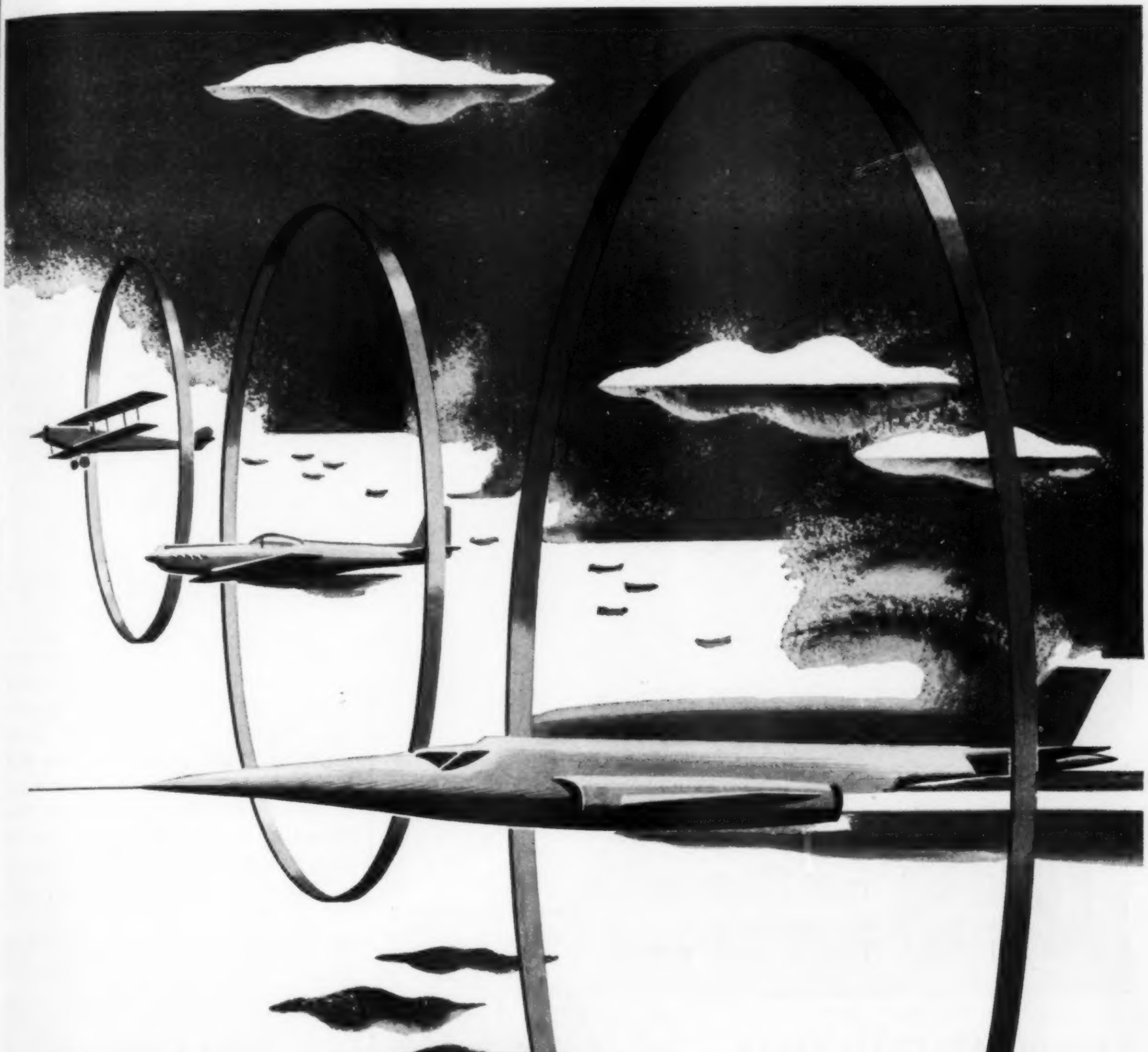
precision optical system includes special reflectionized or opaque 17 × 17-in. screen and luminized mirror. Because of the projection of the image directly on its screen, as well as high light concentration provided by a 500-w lamp, the machine may be used to make office enlargements from microfilm when used with modern photocopy machines. It is mounted on a table which provides storage space and portability. Made by **Filmsort Inc.**, 50 S. Pearl St., Pearl River, N. Y.

For more data circle MD-111, Page 211

Amplifier System

Unitized, multiple-channel amplifier System D now offers extended carrier-amplifier range and increased data capacity. Both linear-integrating and carrier amplifier units may be used in any combination up to a maximum of 12 to permit simultaneous use of self-generating and externally excited pickups. Physical phenomena in the range of 0 to 5000 cycles per second may be amplified and system output coupled to any recording oscillograph equipped with galvanometers of matched characteristics. The system is used with capacitance, inductance, reluctance, resistance, piezoelectric, photoelectric and special transducers. Operating temperature range is −10 to 40 C ambient. Carrier amplifiers provide uniform frequency response from input signals in the 0 to 600 cycles per second range. Full-scale output may be obtained from input



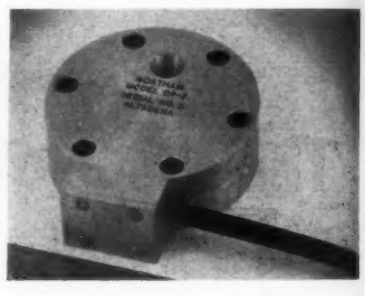


signals ranging from 1 mv to 1 v. Flat frequency response of the linear-integrating amplifiers is as follows: linear, ± 5 per cent from 5 to 5000 cycles per second; integrating, ± 6 per cent from 5 to 5000 cycles per second (low range) and from 50 to 5000 cycles per second (high range). Input signals for linear amplification may be as low as 3 mv; those required for integrating operation are 190 mv in low range and 19 mv in high range. Made by **Consolidated Engineering Corp.**, 300 N. Sierra Madre Villa, Pasadena 15, Calif.

For more data circle MD-112, Page 211

Pressure Transducer

Model DP-7 sensitive pressure transducer measures differential pressure in terms of electrical output. The range of working fluids includes all non-corrosive gases and liquids. A selection of working ranges from ± 0.5 to ± 15 psi differential at line pressures up to 100 psi is offered. Operation is



based on the variable reluctance principle. Absence of mechanical loading on the pressure sensing diaphragm results in a high natural frequency, necessary for good dynamic response and freedom from mechanical shock excitation. Output is derived from two inductances which vary 10 per cent in opposite directions with application of full pressure. The instrument operates in ac carrier systems from 60 to above 10,000 cycles per second with an accuracy, including nonlinearity and hysteresis effects, of 1 per cent of full scale. Made

Engineering Equipment

by **North American Instruments Inc.,** 2420 N. Lake Ave., Altadena, Calif.

For more data circle MD-113, Page 211

Template Service

Custom-made plastic templates offering shapes and forms not available on standard templates are designed to save drafting time. Made according to specifications from original blueprints, the plastic templates are individually cut in any quantity from 0.030 clear plastic with a 15-degree bevel for the tracing pencil. Made by **Hermes Engravers Inc.,** 13-19 University Place, New York 3, N. Y.

For more data circle MD-114, Page 211

Vacuum Gage

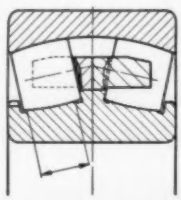
Model 516 Pirani vacuum gage operates in the pressure range between 1 and 0.001mm of mercury (1000 to 1 microns). The gage's low impedance power rectifier eliminates the need for voltage adjustments during operation. One meter is used for both voltage adjustment and pressure reading. A compensating element mounted inside the cabinet facilitates installation and transportation of the gage. Circuitry on the pressure range selector switch permits outgassing at any pressure without danger to the filament. Gage operates on 60 cycles, 110 v ac. Made by **National Research Corp.,** 70 Memorial Dr., Cambridge 42, Mass.

For more data circle MD-115, Page 211

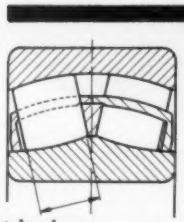
the SKF

improvement of internal design
of spherical roller bearings

Increases capacity 25% to 50%
increases service life 2 to 3½ times



This is the Spherical Roller Bearing design originated by SKF *more than 30 years ago*. The cross-section shows how the integral inner ring flanges, and the undercuts adjacent thereto, limit the effective length of the rollers.



Here is the latest SKF improvement, a *revolutionary advance in design*. Effective roller guiding is accomplished by means of a separate ring. The need for undercuts is eliminated! This type of guide ring permits the rollers to take the position which their contact with the rings dictates. This assures uniform load distribution over the entire length of the longer rollers at all times. *Result*—greatly increased capacity and life.

Now, SKF, originator of the Spherical Roller Bearing, has again provided industry with another anti-friction *first*.

This time, it's a revolutionary design advance in Series 222 and 223 bearings providing these advantages:

When designing new equipment, you can obtain the desired life by using smaller or lighter bearings at considerable cost saving. In addition, this forward step in design will carry heavier combinations of radial and thrust loads, or pure thrust loads of greater magnitude. The important self-aligning feature of SKF Spherical Roller Bearings has been preserved, so that considerable misalignment between the shaft and housing has no ill effect whatever on bearing capacity or life.

During more than 4 years, thousands of installations have been made in railroad journals, vibrating screens, steel mill machinery and numerous other fields. In all cases performance has been outstanding.

SKF Sales Engineers in our District Offices throughout the country will assist you in making use of the important advantages of the improved internal design of SKF Spherical Roller Bearings. **SKF INDUSTRIES, INC.,** Dept. 616, PHILADELPHIA 32, PA. —*manufacturers of* SKF *and* HESS-BRIGHT *bearings.* 7424

This 12-page booklet gives you additional facts—sizes available—added capacity, size by size—increased life you can expect for each size—dimensional tabulations—and load and speed data. Write now for your free copy of Booklet No. 365-2.

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BALL AND ROLLER BEARINGS

Stress Relief

SOME time ago in this column, J. P. Henderson began a series of analyses of bosses he's known. Here's one known to many.

Mr. Big

Included on that list of bosses I would consider inadequate is a former employer of mine. I worked for him for some years, and later when we both changed jobs, my business brought me into frequent contact with him and his department.

In a way I suppose this is a character sketch of the man, although I will keep it on the employer-employee relation. I'll call him Mr. Big for reasons which will become obvious.

Mr. Big died several years ago and various engineering organizations struck off resolutions full of "whereas'es." Eulogies appeared in numerous places.

I had worked for him years ago at the University where he had been a department head. He seemed well along in years then, and in fact he was old enough to belong to that period when there were few college-trained engineers. Almost all of those graduates became famous or so it seemed. At any rate, those that did all appeared to rate the title of "pioneer" in this or "pioneer" in that.

He knew them all. They visited him at the University or saw him at conventions.

I wished I might have met some of those men, and so would the other members of Mr. Big's staff. None of us ever did! At that time I was young enough to have been impressed with big names. It became obvious to us in his department that we were all underlings, of no technical nor social importance. It gradually became obvious too, that he hired us that way. He never seemed to recruit for his department any big names nor even rising young men who appeared to be going somewhere. Perhaps they might dim his luster, or

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Reliance Cold Finished Steel is available in rounds, squares, flats or special shape sections in wire sizes 020 x 020 x ½ to 5/16 in coils or cut to length. Write direct for information on sizes, shapes and analyses.

Reliance Hoz-Fas-Ners save time and money wherever rubber, plastic or fabric hoses are used. Rust resistant, re-usable, one piece design . . . maintain uniform automatic tension. Write for Engineering Bulletin No. 1.

Reliance Snap Rings save man hours and material on shaft assemblies, bearing applications and in counterbore assemblies. Shoulders quickly formed by machining a groove and snapping in a Reliance ring. Send for Bulletin 4K/3.

Eaton Thread-Cutting Springtites and Sems eliminate tapping when fabricating products. They cut their own threads as they fasten assuring perfectly mated, tightly seated threads. Full information in Engineering Bulletin S-49A.

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make him appear less obviously head and shoulders above his staff.

I recall one interview very distinctly. As an ambitious young cub on the teaching staff I wanted to do some development work and write a paper or two. I was inexperienced enough to mistrust my own judgment as to suitable fields to tackle and I resolved to ask his advice. Now a professor has been defined as a man who can say "yes and no" in ten thousand well chosen words. Mr. Big gave me exactly that kind of advice leaving me with the impression that I was to be encouraged, but still there was no tangible help from the interview.

Shortly after, I joined that rather large group of men who *had* worked for Mr. Big.

I was amazed some years later to learn that he too had left the University. He was fairly old for such a basic change (or so he appeared at the time) and he had been considered a University fixture. I never knew whether his inadequacies had caught up with him or whether he had left voluntarily.

He became head of a development department in an industrial concern, and after some years my job put me in frequent contact with him and his organization. History repeated itself, for as I became better acquainted with one of the older members of the department I found rumblings of dissatisfaction. Aloof, never told anyone anything, thought himself better than anyone else—those were the comments.

As a boss, Mr. Big belonged back in the old feudal system. In effect he was the lord of the manor, and anyone who worked for him was a serf, always to be treated politely and with reasonable kindliness, but fundamentally of no importance. And one of the things that marked him was a serenity that no employee of his ever seemed to ruffle. Perhaps that's why he lived so long!

"Inspiring teacher, molder of men, an inspiration to hundreds of technical men who came under his guidance. . ."

Well, that's what his eulogy said!

—J. P. Henderson

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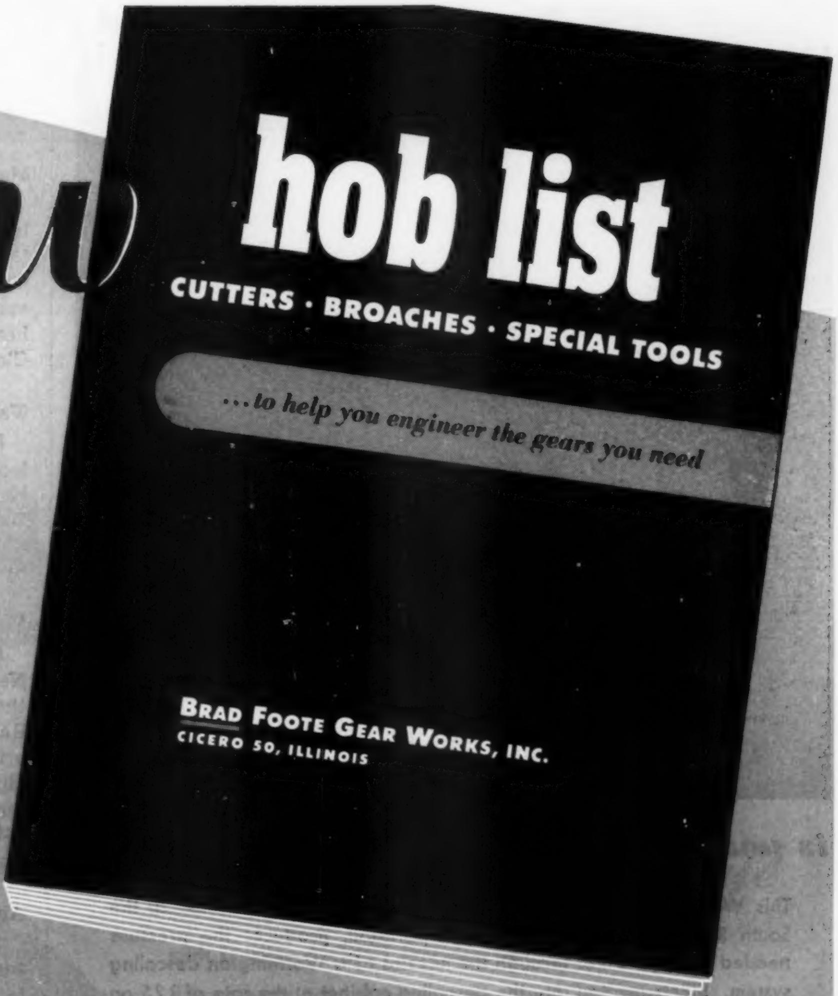

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Meetings

AND EXPOSITIONS

Mar. 16-17—

Steel Founders' Society of America. Annual meeting to be held at the Edgewater Beach Hotel, Chicago, Ill. Additional information may be obtained from society headquarters, 920 Midland Bldg., Cleveland, O.

Mar. 17-19—

Pressed Metal Institute. Spring technical meeting will be held at Hotel Carter, Cleveland, O. O. B. Werntz, 2860 East 130th St., Cleveland, O., is managing director.

Mar. 22-25—

Institute of Radio Engineers. National convention to be held at the Waldorf-Astoria Hotel and the Radio Engineering Show at Kingsbridge Armory in the Bronx. Additional information may be obtained from society headquarters, 1475 Broadway, Times Bldg., New York 36, N. Y.

Apr. 3-4—

Packaging Machinery Manufacturers Institute. Spring meeting to be held at Hotel Dennis, Atlantic City, N. J. Additional information may be obtained from society headquarters, 342 Madison Ave., New York 17, N. Y.

Apr. 5-7—

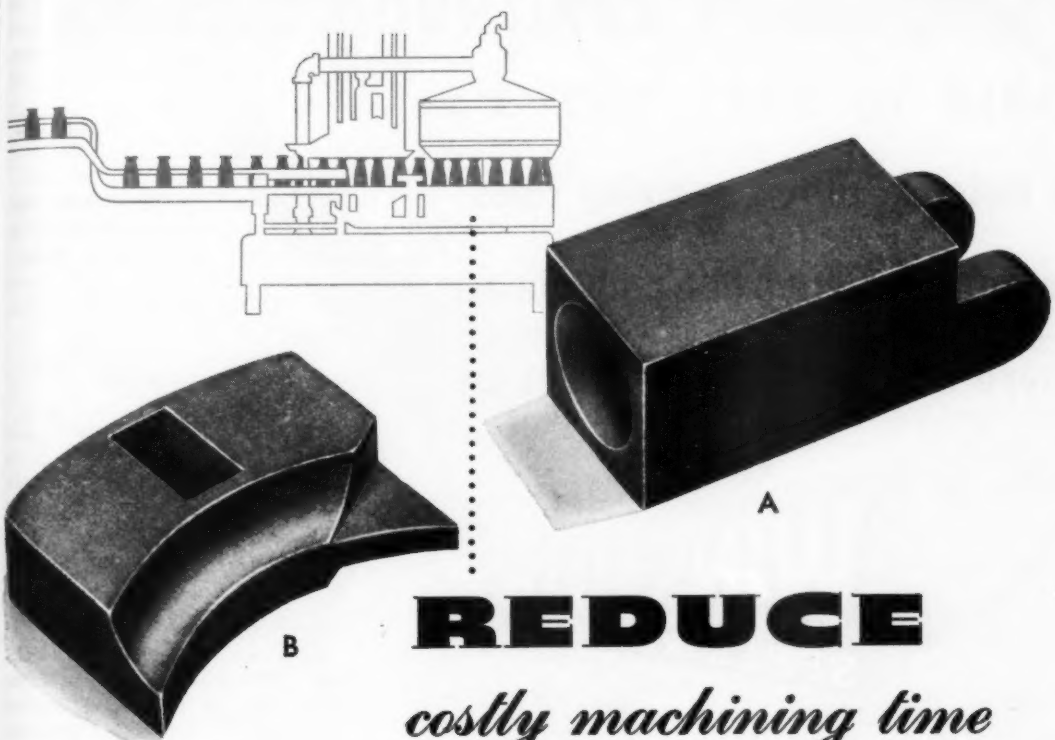
National Fluid Power Association. Spring meeting to be held at the Edgewater Gulf Hotel, Edgewater Park, Miss. Barrett Rogers, 1618 Orrington Ave., Evanston, Ill., is executive secretary.

Apr. 5-7—

American Society of Lubrication Engineers. Annual meeting and exhibit to be held at the Hotel Netherland Plaza, Cincinnati, O. Additional information may be obtained from society headquarters, 84 East Randolph St., Chicago 1, Ill.

Apr. 14-16—

Society for Experimental Stress



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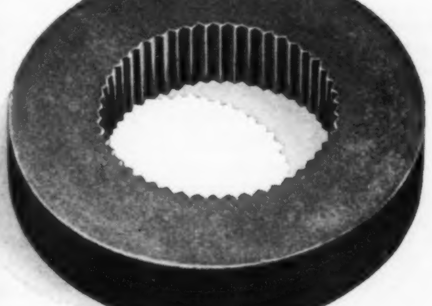

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Meetings and Expositions

Analysis. Spring meeting to be held at the Hotel Netherland Plaza, Cincinnati, O. W. M. Murray, P. O. Box 168, Cambridge 39, Mass., is secretary.

Apr. 18-23—
American Ceramic Society Inc. Fifty-sixth annual meeting to be held at the Palmer House, Chicago, Ill. Charles S. Pearce, 2525 North High St., Columbus 2, O., is secretary.

Apr. 20-21—
American Zinc Institute Inc. Thirty-sixth annual meeting to be held at the Hotel Statler, St. Louis, Mo. Ernest V. Gent, 60 East 42nd St., New York 17, N. Y., is secretary.

Apr. 26-28—
American Management Association. Manufacturing conference to be held at the Hotel Statler, Cleveland, O. Additional information may be obtained from society headquarters, 330 West 42nd St., New York 36, N. Y.

Apr. 26-28—
Metal Powder Association. Annual meeting and exhibit to be held at the Drake Hotel, Chicago, Ill. Additional information may be obtained from society headquarters, 420 Lexington Ave., New York 17, N. Y.

Apr. 26-30—
American Society of Tool Engineers. Convention and industrial exposition to be held at Bellevue-Stratford Hotel, Convention Hall and Commercial Museum, Philadelphia, Pa. H. E. Conrad, 10700 Puritan Ave., Detroit 21, Mich., is executive secretary.

May 4-8—
American Welding Society. National spring technical meeting and welding and allied industry exposition to be held at Hotel Statler and Memorial Auditorium, Buffalo, N. Y. J. G. Magrath, 33 West 39th St., New York 18, N. Y., is secretary.

By A. F. Welsh
Manager
Worthington Corp.
Reciprocating Pump Div.
Harrison, N. J.

DESIGN ABSTRACTS

Heavy-Duty Hydraulic Accumulators

PRIMARY function of a hydraulic accumulator is to supply instantaneous and peak demands for high-pressure liquid with relatively low-capacity pumping equipment. Pump energy is stored for intermittent duty cycles by using the time in between periodic load requirements to replenish the accumulator, *Fig.* 1. In addition, accumulators help control flow through the system and, in the case of the air-loaded type, provide a certain amount of cushioning of hydraulic shock.

There are three basic types of accumulators: weighted, air piston and air bottle. Reference to hydro-pneumatic types has been purposely avoided for clarity, since this name is sometimes applied to air bottles and other units of the air-piston type. The trend today seems to be toward the air bottle, especially for larger sizes.

Weighted Type: The weighted accumulator, often called gravity or ballast type, *Fig.* 2, is a simple device requiring no auxiliaries. It has the advantage that pumps can be controlled by a simple float or limit switches. Release holes in the ram serve to prevent the ram from going too high and tipping over if the controls fail. Another advantage is that when starting the hydraulic system the working pressure of the system can be obtained as soon as the piping is filled and the accumulator is raised off the blocks.

Although the weighted accumulator, by its nature, would seem to provide a uniform pressure, in actual operation there may be a variation of as much as 10 per cent resulting from packing friction on the plunger, and inertia. At times, considerable excess pressure is required to reverse the direction of the falling weighted plunger.

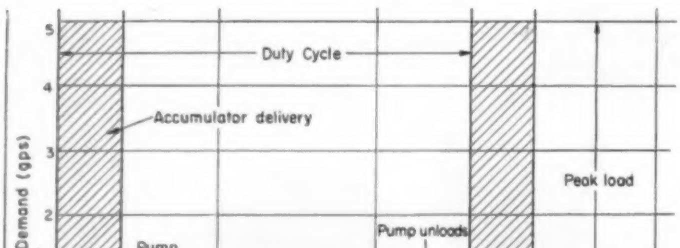


Fig. 1—Above—Supply-demand characteristics for typical hydraulic system utilizing accumulator to meet intermittent duty cycle requirements

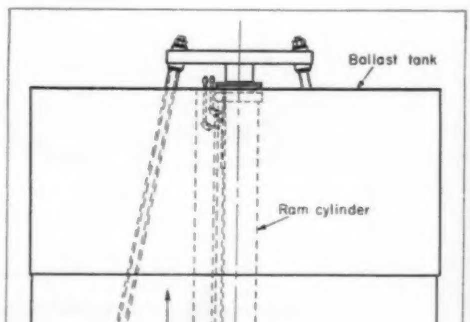


Fig. 2—Right—Weighted or gravity type of accumulator employing ballast tank mounted to ride on roller guides

Air Piston Type: Also known as the intensifier type, this accumulator design, *Fig.* 3, reduced much of the shock caused by the inertia of the ballast weight. In addition, it eliminated the need for a massive foundation. An inexpensive air compressor rated for a maximum of 250 psi can usually supply the air to load the piston. An auxiliary air receiver about 10 times the size of the piston chamber is used to provide sufficient volume of air to limit the pressure drop to about 10 per cent.

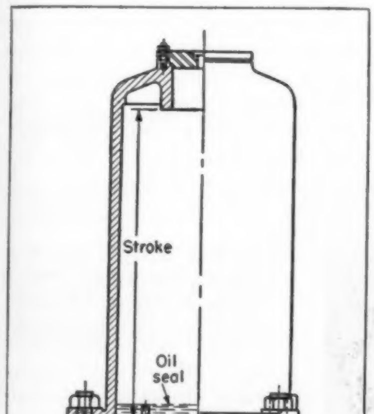
Air Bottle Type: Because of its many advantages, the air bottle is coming into more general use. Its flexibility is practically limitless within the pressure rating of the vessels and the charging compressor. Air bottles can be used singly or in multiples, and can be operated in parallel, series or a combination of both. Pressure differentials as low as 1 or 2 per cent could be maintained if necessary; however, this would require an abnormally high total volume in relation to usable accumulator capacity and would necessitate sensitive instruments for control. Also, the pumping equipment would probably load and unload more often than would be considered good practice.

Standard operating-pressure differential is usually in the neighborhood of 10 per cent, but will vary slightly with the requirements of the machinery being operated. Should hydraulic requirements increase or decrease, bottles can be added or subtracted with relative ease — if consideration is given when the system is designed. System arrangement should permit bottles to be isolated for inspection.

The air or gas charge in the accumulator makes it a very effective surge chamber for the piping system, minimizing hydraulic shocks and pulsations. Usually the charging media is air, but an inert gas can be used if the hydraulic fluid is combustible. Most large hydraulic systems use water with a small percentage of soluble oil.

During the cycling of an air bottle, it is possible for the compression or expansion to range from

Fig. 3—Below—Air piston type of accumulator



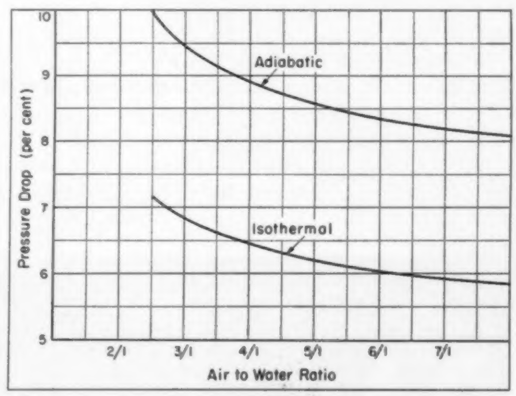
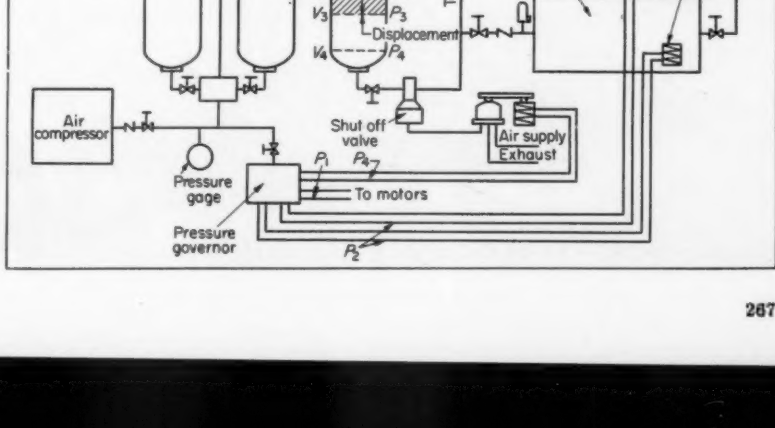


Fig. 4—Relative pressure drop in air-bottle accumulators for different air-water ratios. Total volume is assumed constant

Fig. 5—Below—Schematic layout of typical hydraulic accumulator system showing pressure and volume points including overpressure motor shutdown, P_1; unloading pressure for pumps, P_2; minimum design system working pressure, P_3; accumulator safety shutoff, P_4, and loading pressure for pumps, P_5



isothermal to adiabatic. As a consequence, in estimating vessel sizes a good practice is to assume polytropic conditions, using an n value quite close to adiabatic. This is more conservative than calculations based on isothermal.

A water table of sufficient depth should be provided at the minimum level in the air bottle to prevent loss of the air charge. This depth can vary with service conditions; a liquid demand characteristic causing a high velocity discharge from the bottle requires a larger body of residual water than a long slow draw. To eliminate vortex formation, a baffle over the discharge opening is often used. Large volumes of air escaping from the accumulator into piping and presses can cause serious damage.

Ordinarily, the amount of air that has to be added to a system after it is in operation to compensate for leakage and absorption is relatively minor. In some instances the compressor does not have to run more than 1 or 2 hours a week. Therefore, when selecting a charging compressor, a compromise has to be made in the interests of economy. Often the size of compressor selected is such that 100 to 150 hours of operation will be required for the initial charge. Loss of a large amount of air may require shutting down press operations for a number of hours until the bottles can be recharged; consequently, the cost of down time should be evaluated in the selection of a compressor.

To prevent loss of water and air charge from the bottle in the event of a line break or a valve sticking open, an automatic safety shutoff valve is usually installed near the accumulator outlet. This valve is operated by a low-level or pressure switch, and stops all flow from the accumulator if the level nears the danger point.

Ratio between the volume of air and water in air-bottle systems varies considerably — anywhere from 2 parts air to 1 part water to 8 parts air to 1 part water. As indicated by *Fig.* 4, the greater air-water ratio provides a smaller pressure drop per gallon of water withdrawn. It will be noted that there is a considerable change in pressure depending on whether the expansion or compression is adiabatic or isothermal.

Larger systems can usually use a higher air-water ratio. The minimum volume of water necessary to prevent air loss is a smaller proportion than it would be in a smaller system having the same discharge velocity. Multiple-bottle systems in series operation lend themselves to the use of a minimum residual-water volume. The proportionately smaller diameter of the multiple bottle contrasted to a single bottle of the same total volume and height, gives a thicker bed of water for the same volume.

Pressure and liquid level can be maintained in an air-bottle accumulator by means of pressure governors or level controls, or a combination of both. Control requirements vary in most instances; the controls load and unload the pumps, shut down the pumps in the event of overpressure or high level and

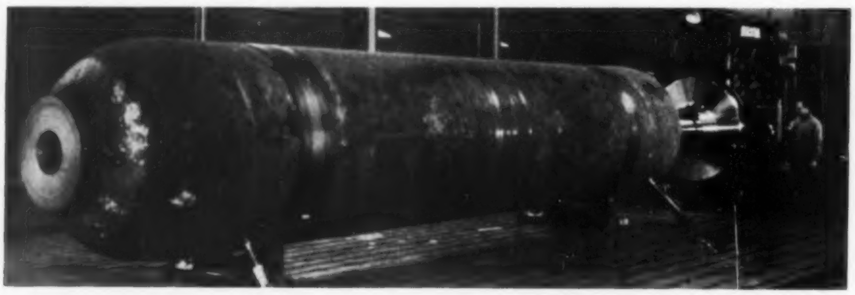


Fig. 6—Left—Forged bottle for a 50,000-ton Mesta press hydraulic system

Fig. 7—Below — High-capacity air-bottle accumulator station for supplying the combined hydraulic requirements of two Loewy presses rated individually at 35,000 and 50,000 tons

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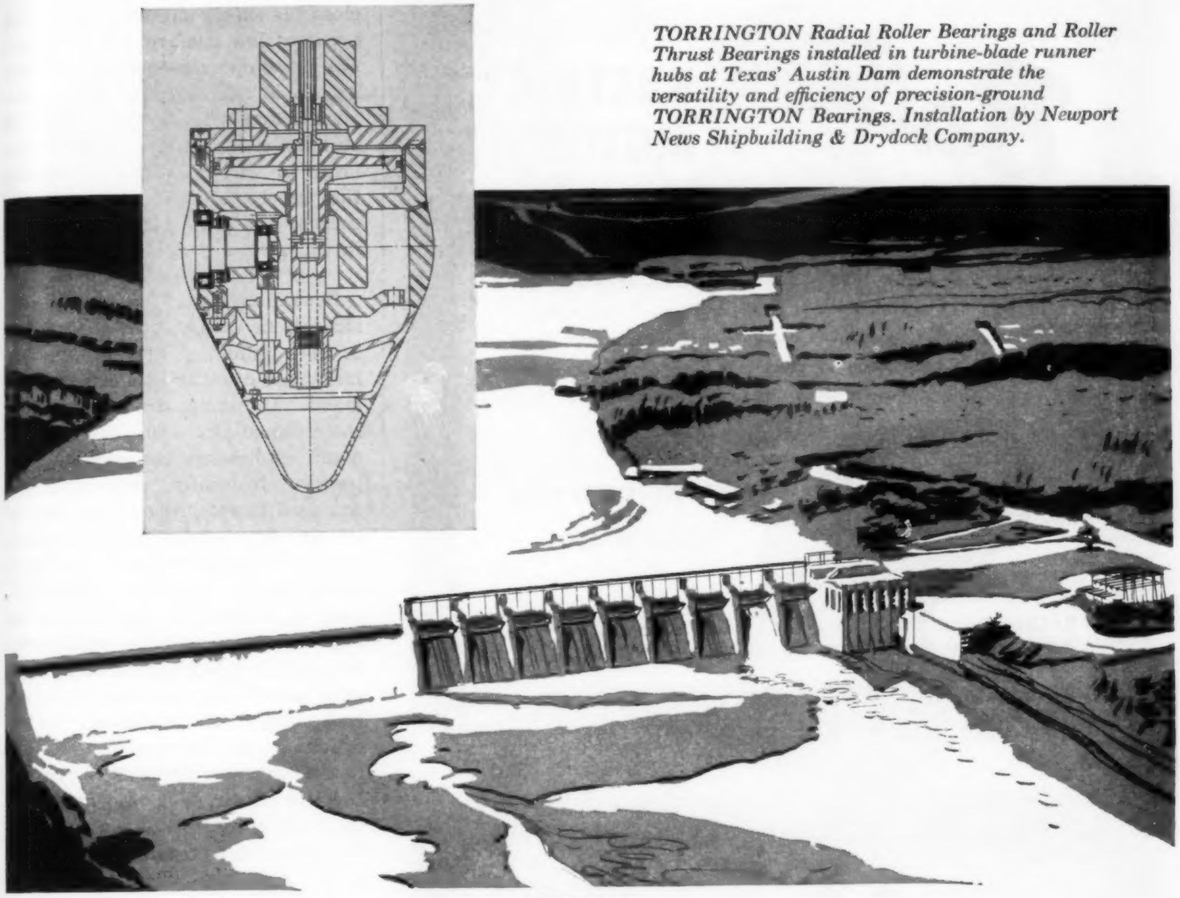

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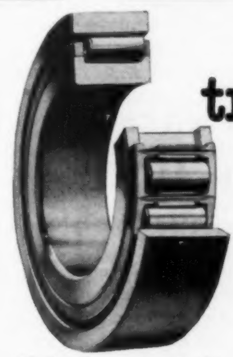

The two Newport News adjustable blade turbines installed at Texas' Austin Dam in 1939 were equipped with TORRINGTON Bearings in the runner hubs. Today, *fifteen years later*, those bearings are still giving efficient, trouble-free service.

And little wonder. For TORRINGTON Bearings are precision ground from the finest of bearing-quality steels. They are carefully heat treated by the most advanced metallurgical methods. In most designs their one-piece cages are cast of bronze and are *land-riding*.

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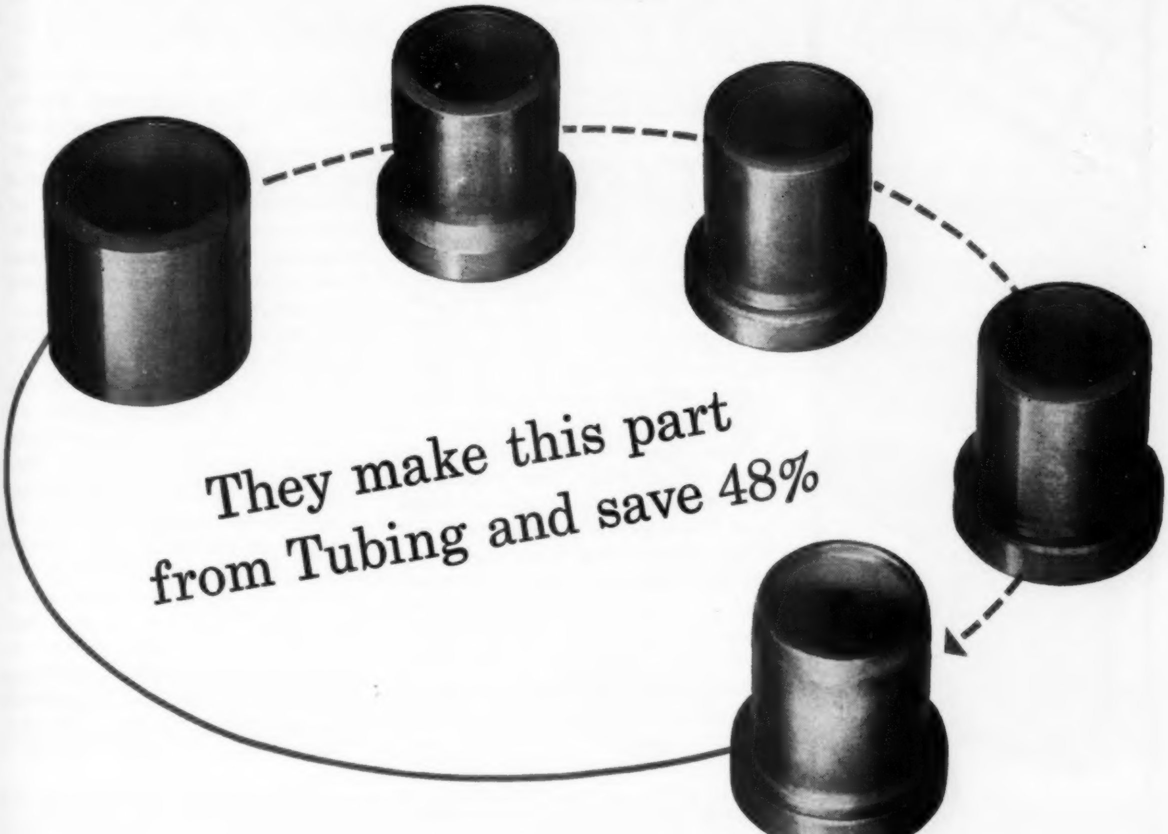
close the safety shutoff valve in the event of low level or low pressure. For press operations such as piercing, it is necessary that the low-level alarm sound when there is still sufficient water in the accumulator to withdraw the die before the safety shutoff valve closes, thereby preventing its being frozen in the metal.

System Design: Two major considerations in sizing an accumulator and pumping system are the maximum draw and the average demand. Maximum draw determines the accumulator size, average demand establishes the pumping capacity. Normally, two capacities are used in referring to the size of an air bottle: total volume and effective or usable volume. In *Fig.* 5, usable volume is shown between V_2 and V_3. This is the volume of liquid which can be withdrawn without dropping the working pressure below the minimum differential pressure of the system. In addition to the usable water and the minimum water required to prevent the loss of air, it is also necessary to have sufficient water to allow the operation of alarms and low-water controls. This amount is shown between V_3 and V_4.

Large air bottles being manufactured today can be of forged, solid welded, or laminated construction. Usually the type of construction selected is based on economic considerations. A forged bottle for a 50,000-ton press is shown in *Fig.* 6.

For large high-pressure bottle charging, multistage air compressors are used. Common practice is to start these compressors manually; however, they are often arranged to stop automatically when the pressure reaches the maximum working value.

Centralized Stations: An interesting application of air-bottle accumulators to meet extreme hydraulic requirements is shown in *Fig.* 7. Designed to meet the combined hydraulic requirements of two presses rated at 35,000 and 50,000 tons, this centralized accumulator station will actually employ seven triplex pumps developing a total of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Design Abstracts

10,500 horsepower. All of the major components of the high-pressure system, as well as the prefill vessels, are shown. The accumulators each have a capacity of 350 cubic feet and are 36 feet tall. Seven bottles are required for the 50,000-ton press and four for the 35,000-ton unit. Four pumps supply the large press and three the smaller. A spare has also been provided for each press. It is possible for the presses to be operated separately from either accumulator unit in case of servicing.

The "master" water bottle contains a float which passes through the field of two induction coils mounted externally. As the water level drops during press operation, the float enters the field of the "cut-in" or "loading" coil. The float acts as a core in a regular induction coil, actuating a time-delay switch which allows the pumps to cut in consecutively with an adjustable and predetermined time lag. This method of control prevents sudden large load conditions at the power station. Restoration of the water level reverses the action.

Today, facilities are available to completely shop fabricate an air bottle having a volume of 11,800 gal, or possibly greater, and suitable for a pressure of 4500 psi. This fact, together with recent pump developments and the possibility of using multiple units, will allow the design of accumulator stations in sizes far beyond any contemplated, even in connection with the present Air Force program.

From a paper entitled "Capacities and Sizes of Equipment in Accumulator Stations for Heavy-Press Operation" presented at the Annual Meeting of ASME in New York, N. Y., December, 1953.

Pearlitic Malleable Iron

By Carl F. Joseph

Central Foundry Div.
General Motors Corp.
Detroit, Mich.

PRODUCTION of pearlitic malleable iron castings is carried out in much the same manner as is the manufacture of malleable iron castings. The major difference lies in the heat treatment cycle of white iron. The base metal used is generally of the same chemical composition as that of a regular malleable iron casting. However, some manufacturers make additions of manganese, copper manganese, molybdenum and other elements. These additions control or regulate the graphitizing rate during second stage graphitization of this material.

The heat treating cycle in the production of pearlitic malleable iron, in most cases, is different than that used for malleable iron. The matrix in a malleable iron casting is completely annealed, resulting in a structure of ferrite and temper carbon. Some manufacturers reheat regular malleable iron to above the critical, liquid quench and temper. Where alloys are added to the base metal, the regular malleable annealing cycle can be successfully employed.

Mechanical Properties

Pearlitic malleable iron has many properties which make it outstanding in the ferrous field. Its properties differ, depending on the amount of combined carbon in the matrix. The higher the combined carbon, the stronger, harder and less ductile this type of iron becomes. The control of the percentage of combined carbon offers the user of pearlitic malleable iron a wide range of mechanical properties.

Machinability: In general, the machinability of pearlitic malleable iron is from 10 to 30 per cent better than steel forgings of the same Brinell hardness. Tool life is substantially increased due to easier machinability and the reduction of excess finish stock.

Bearing Properties: This type of iron exhibits excellent nonseizing properties in metal-to-metal wear. For example, in the design of an automotive rocker arm which operates on a hardened steel shaft, the bronze bushing was eliminated.

Finishing Characteristics: The metal can be machined to a very

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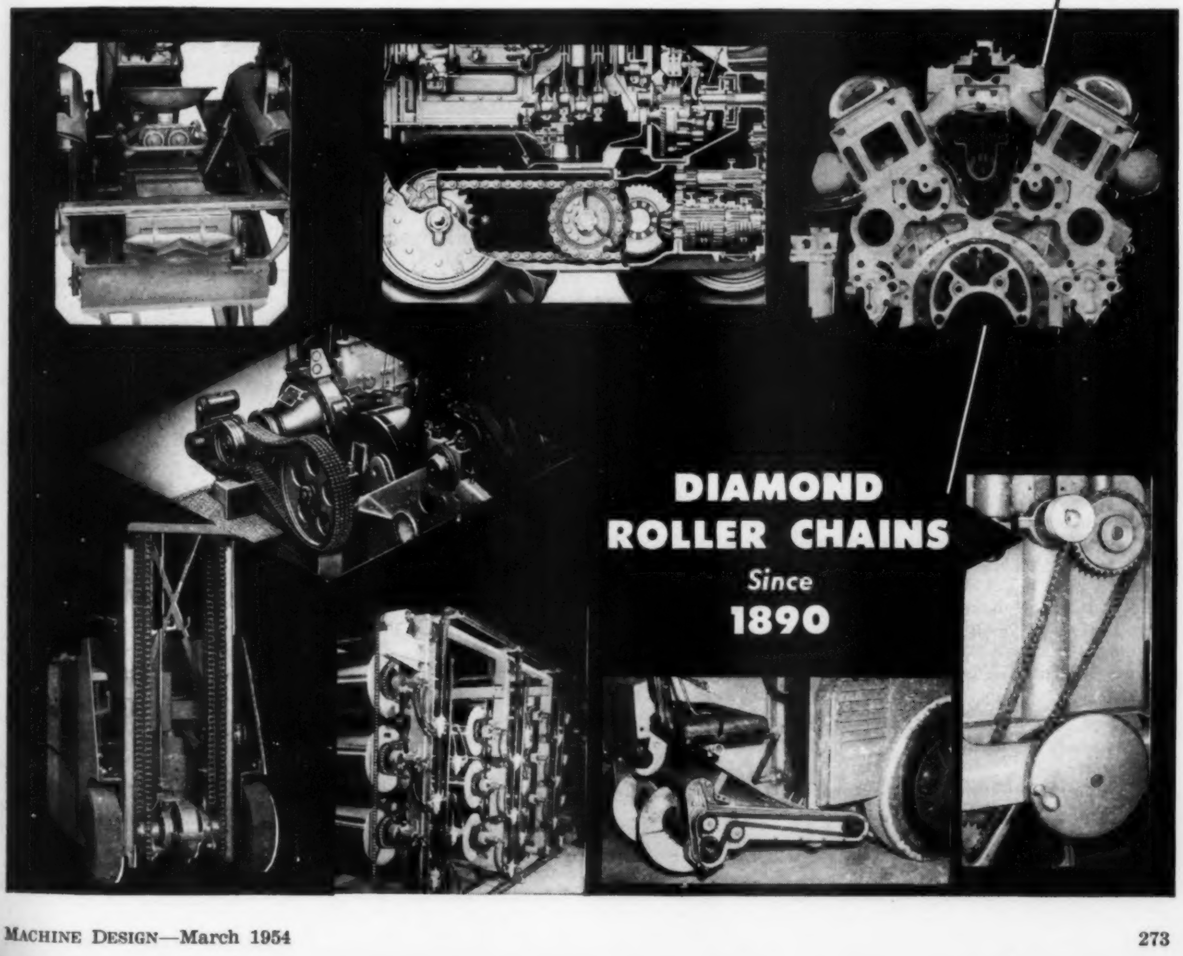

Design Abstracts

smooth surface and polished to a mirror-like finish thus reducing friction and wear to a minimum.

Strength Properties: Possessing minimum yield strengths of 80,000 psi and an ultimate strength of 100,000 psi, pearlitic malleable iron castings can be used for many highly stressed parts. The matching of high yield strength with good machinability is a rare combination.

Hardening Properties: Pearlitic malleable iron responds readily to localized hardening by either the flame hardening or induction hardening method. This operation is used on shifter yokes, crankshafts and gears. Immersion in a lead or salt bath is employed on parts which can be immersed in the hot molten bath followed by an oil quench. Liquid bath hardening can also be applied on lower hardness pearlitic malleable iron by allowing a longer time at a higher temperature. This produces enough combined carbon to give a minimum Rockwell C of 50 with oil quenching.

Castings illustrating selective hardening include trolley wheels, rocker arms, truck crankshaft sprockets, some small engine crankshafts, transmission shifter yokes and many others.

Fatigue and Wear Resistance: Pearlitic malleable iron provides good resistance to fatigue, giving maximum endurance and long life. Castings offering advantages in reducing costs and contributing to economical production are the parts which must have good wear resistance. Pearlitic malleable iron withstands excessive wear under heavy loads at high speed.

Typical applications in the automotive field include automatic transmission gears, transmission pistons and, in general, parts which are produced in the Brinell range in excess of 241. These parts can also be classed as high-strength castings such as universal joint yokes, transmission shifter yokes and certain compressor crankshafts.

From a paper entitled "Pearlitic Malleable Iron—Its Properties and

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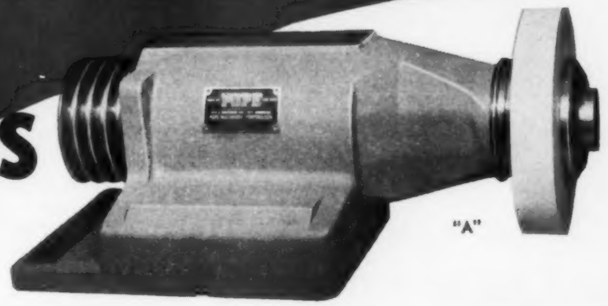
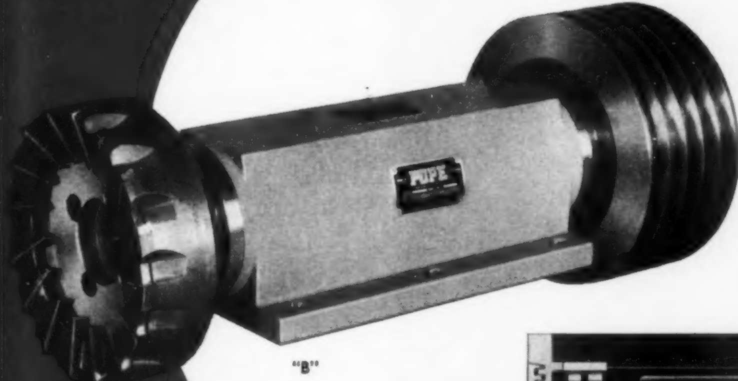
HEAVY DUTY WHEEL HEADS



"A"

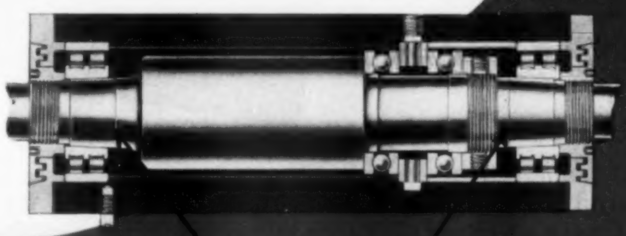


"B"

Note the big precision bearings and the big shaft.

A pair of precision thrust bearings stabilize the shaft against any axial movement in either direction. (No end play)

The double row cylindrical roller bearings carry the radial loads with ease and at low temperature.



Taper mounting provides for precise individual preloading of the radial bearings.

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Design Abstracts

Expanded Uses" presented at the SAE Annual Meeting in Detroit, Mich., January, 1954.

Design Factors for Journal Bearings

By P. P. Love, P. G. Forrester and A. E. Burke

Glacier Metal Co. Ltd. Wembley, England

FACTORS in the operation of bearings may be classified according to whether they are inherent in, accessory to, or derive from the functional requirements of the bearing. The first group is predetermined; the second group is conditional; and the third group is by design. They are described accordingly as:

1. Primary or functional factors. Forces to be transmitted—magnitude, range, direction, and distribution—and relative motion between elements — magnitude and range.
2. Accessory factors. Ambient conditions such as heat, oxygen, etc., oil (physical and chemical properties and conditions of supply) and structural properties (thermal and mechanical).
3. Derived factors. Journal surface —material (physical, chemical and mechanical properties), geometry (contour, surface finish); bearing liner surface—material (physical, chemical, and mechanical properties), geometry (contour, surface finish); and journal and housing—structural properties (deflection under load).

Distribution of the load over the bearing is a major, if not the most significant, factor and, as will be seen, depends wholly upon accessory and derived factors.

Coefficient of Friction: The manner in which oil affects the operation of plain bearings is complex and only the simpler conceptions have been analyzed and formulated. They may be summarized with reference to the well-known ZN/P diagram shown in *Fig.* 1, which illustrates some of the interactions. For constant oil properties Z and constant load P, the coefficient of

This walking beam for a heavy duty machinery trailer originally was produced as a weldment.

A review of the design by the manufacturer's engineering department and the steel foundry engineer showed that the part could be produced more advantageously as a steel casting. Conversion resulted in a reduction in cost of 30% and a reduction in weight of 23%. Appearance also was improved.

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friction μ in a bearing varies widely over a range of speed N. The region C to D and to some extent beyond, towards E, is that region in which an oil film between journal and bearing liner surfaces carries the load so effectively as to prevent contact between the two surfaces. The oil in this range of operation distributes the force to be transmitted by the bearing, and consequently enhances the load carrying capacity.

For constant loads and speeds in the region C to D, there are approximate mathematical formulations which relate load, speed, journal diameter, bearing width, clearance and viscosity. For fluctuating loads, however, there is little precise knowledge in a form which can be used by the designer.

Trapped Volume Effect: The region from B to C is less well understood than C to D, but a variety of mechanisms can be conceived. Because reduction of surface asperities and of particle size in the oil can bring C nearer to the origin, it is considered that the increase of

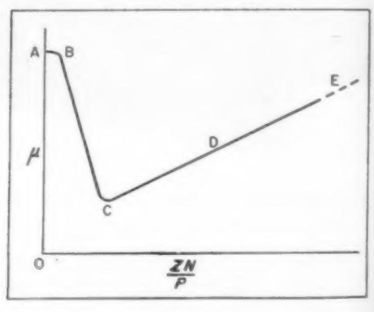

Fig. 1 — Typical ZN/P diagram showing effect of interaction of oil properties, Z, load, P, and speed, N, on coefficient of friction, μ

μ from C to B is due to interference of solids. Except in the direction of point B, however, there is still a substantial degree of oil film or hydrodynamic lubrication. It is conceivable that there is, in addition, a "trapped volume" mechanism of lubrication which is attributable to the existence of surface asperities. This phenomena is shown in *Fig.* 2, which represents

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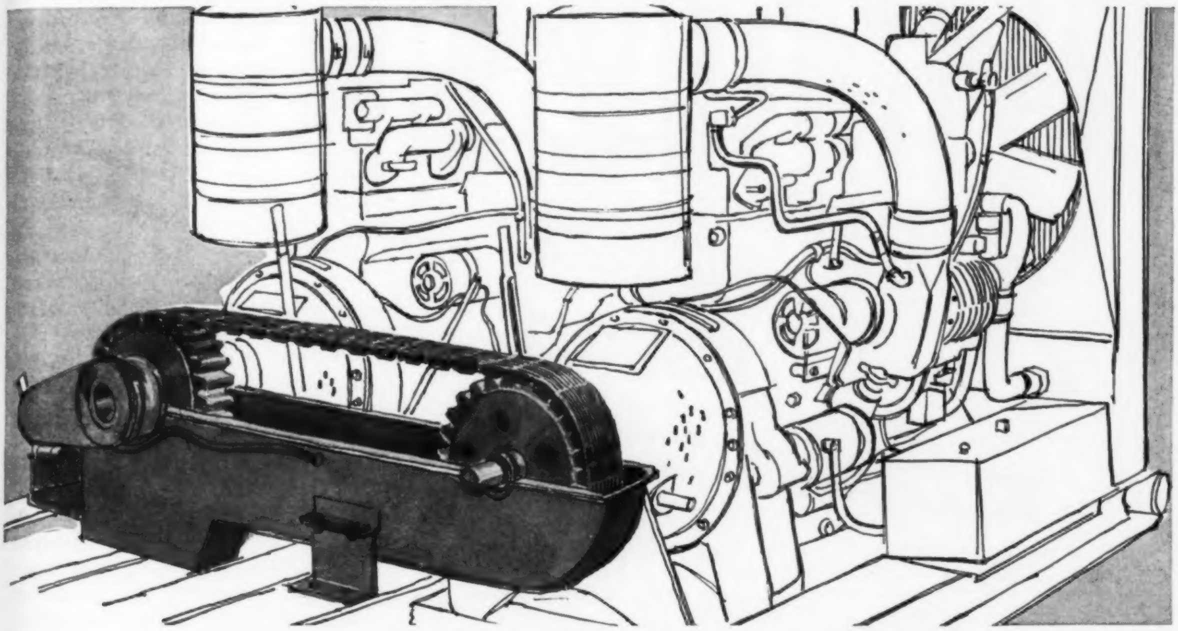

How compounding with HY-VO could save you as much as $4,560

Now, gain new advantages in buying and compounding smaller, higher speed engines with Morse Hy-Vo Drive.

You make major cost savings.

The total cost of two high-speed engines and the Hy-Vo Drives with which they are compounded will usually be considerably less than a single large engine of equivalent horsepower. A brief look at some figures will show how this works . . .

For example, if you needed 250 continuous horsepower at 1200 rpm., you could: 1) Buy a manufacturer's 250 hp., 1200 rpm. engine, which lists for $13,380; 2) buy the 250 hp., 1800 rpm. engine for $11,930 (using Hy-Vo as a connection and reduction drive, also eliminates the costly flexible coupling normally required in choice No. 1); 3) or, use two 130 hp., 1800 rpm. engines, listing for a total of $8,820, and use Hy-Vo as your compounding drive.

With the 250 hp., 1800 rpm. engine, you save $1,450, more than enough to pay for your Hy-Vo Drive. Compounding two 130 hp., 1800 rpm. engines, you save $4,560 over base list cost of the 250 hp., 1200 rpm. engine.

From this saving you could pay for your Hy-Vo Drive and bank several thousand dollars. In most cases, too, Hy-Vo will be so narrow it will eliminate costs of outboard bearings and mountings, with the bonus that single-engine operation is possible under reduced power requirement.

Other power transmission methods generally have either inherent restrictions on rotative speeds and velocities that limit 200 hp. or larger engines to 1200 rpm., or are too costly. Hy-Vo has far greater upper limits, with relatively lower cost.

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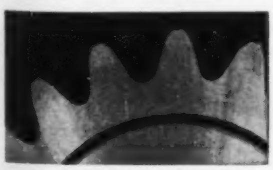
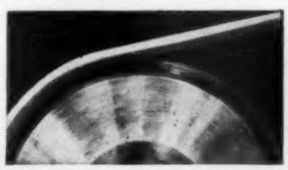


Hy-Vo's many design and manufacturing refinements result in a drive with unusual power transmitting capacity. Note involute tooth design of Hy-Vo Sprockets . . .

. . . and the new, Hy-Vo Compensating Rolling Joint, which always shifts the pitch line automatically as the Hy-Vo Chain engages the involute teeth of the sprocket.



Chain follows a path truly tangent to sprocket pitch circle, virtually eliminating chordal action. A new, balanced link design greatly improves stress distribution.

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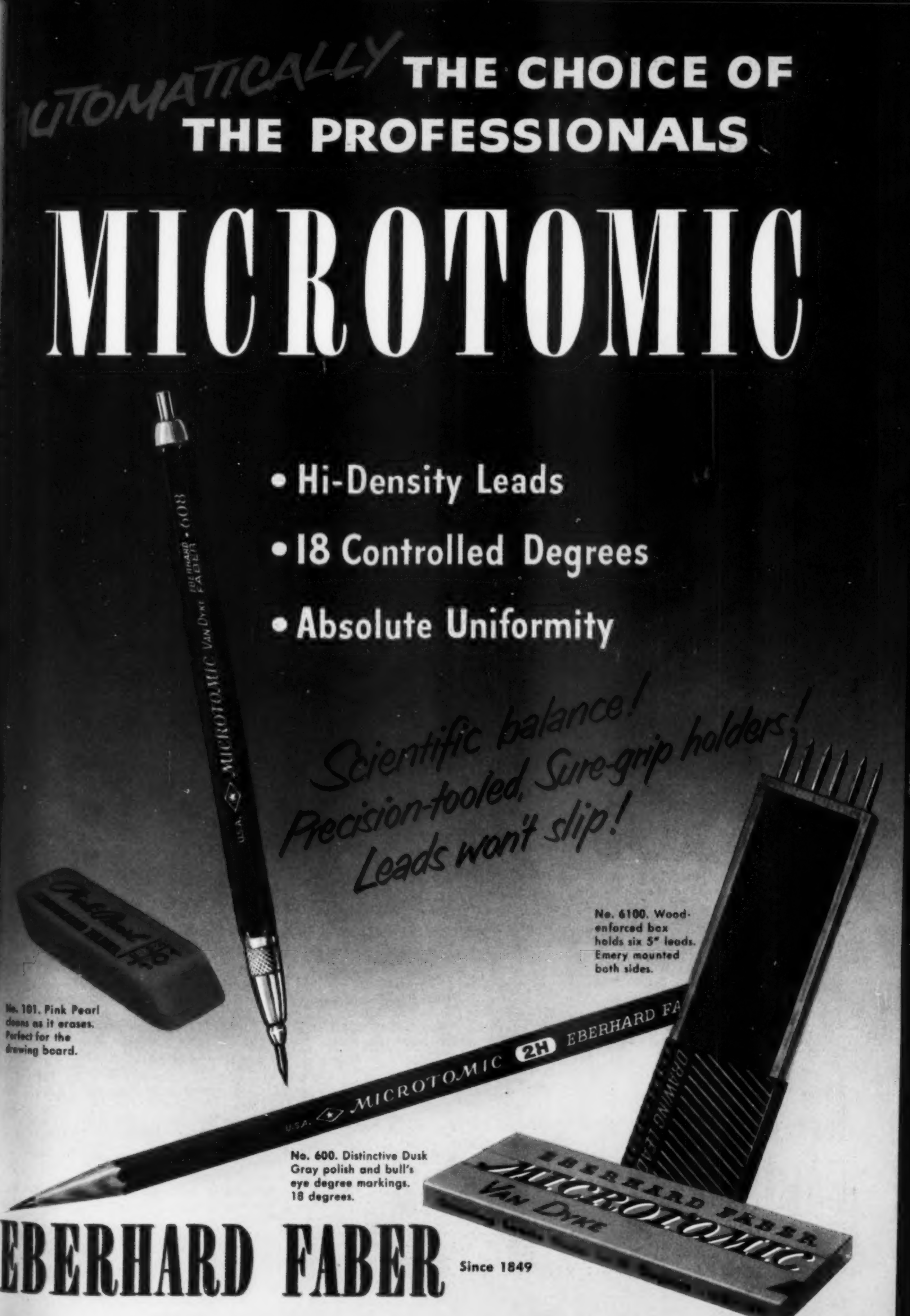
MECHANICAL POWER TRANSMISSION PRODUCTS

on a large scale an elementary section across the "clearance" space between the journal and bearing surfaces. In relation to the bearing, the journal is moving in the direction of the arrow. A volume a bounded by minute clearance or surface contact bands, c, is, by virtue of the particular contours of surfaces shown, reducing in volume with the motion of the journal. Because of the clearance bands c, escape of the oil in a is restricted and a local pressure in a is set up. A volume b, bounded likewise but increasing in volume, is unable to be filled rapidly enough with oil, and a sub-ambient pressure exists. This condition is limited, whereas the positive pressure is not limited and the balance may go to support load between the journal and bearing. It is of interest to consider that in the space b cavitation or deaeration may occur. Also in the B to C region *Fig.* 1, the chemical properties of the journal and bearing liner surfaces, can be designed to alleviate μ. The same interaction of properties also determines the position of A.

From the foregoing, it will be seen that the mere presence of a viscous medium (to say nothing of the various additives of a modern oil) can affect or be affected by every one of the factors described.

Surface Geometry: The relative motion between the elements, that is, between the journal and block, determines the geometry of the surfaces within a narrow range. For radial loads, the surfaces must be generated by revolution and the simplest of such surfaces to manufacture is generated by a straight line. Hence, generally speaking, cylindrical surfaces are used. The forces to be transmitted, however, can cause general deflections and surface deformations. The former is, in effect, the tilting of the axes of the journal and block and can give rise to major and potentially serious redistribution of the forces over the load carrying area of the bearing. There are at least three ways in which this action can be mitigated.

The first method, which applies particularly to crank-mounted bear-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ings, is to design the connecting rod so that the axis of the crank end will tilt in the same direction and, if possible, to the same extent as that of the crankpin when subject to the preferred pattern of force distribution. Within the limits of proportionality, the deflections can be concordant. This technique is not in common use, probably because its value is not fully known.

The second method of preventing undue concentrations of force is to profile the surface of the bearing hyperbolically; that is a surface generated by a straight line which is not parallel with the axis of generation. This has the effect of reducing load concentration when only the journal axis is deflected, an effect achieved at the expense of increasing the load concentration when there is little or no journal axis deflection. But as a rule, this increase is insignificant since zero axis slope is generally associated with zero load. Hyperbolic bores, generated by a line (boring tool traverse) about 2 degrees skew to the axis of the bearing, are adopted by certain continental designers.

The third method is to allow local deflections to take place, but this is not usually advisable as secondary stresses are imposed on the lining.

Bearing Performance: Progress in the development of machinery which involves a multitude of factors can only be directed by the study of failure either as experienced in the field or as produced intentionally by test to destruction. By far the majority of bearing failures are, in the authors' experience, due to faulty assembly.

There are, broadly speaking, three kinds of failure; namely,

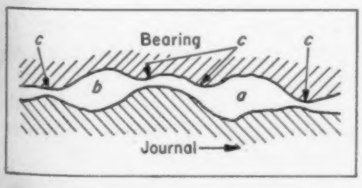

Fig. 2—Schematic interpretation of conditions producing trapped volume lubrication


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Design Abstracts

ings, is to design the connecting rod so that the axis of the crank end will tilt in the same direction and, if possible, to the same extent as that of the crankpin when subject to the preferred pattern of force distribution. Within the limits of proportionality, the deflections can be concordant. This technique is not in common use, probably because its value is not fully known.

The second method of preventing undue concentrations of force is to profile the surface of the bearing hyperbolically; that is a surface generated by a straight line which is not parallel with the axis of generation. This has the effect of reducing load concentration when only the journal axis is deflected, an effect achieved at the expense of increasing the load concentration when there is little or no journal axis deflection. But as a rule, this increase is insignificant since zero axis slope is generally associated with zero load. Hyperbolic bores, generated by a line (boring tool traverse) about 2 degrees skew to the axis of the bearing, are adopted by certain continental designers.

The third method is to allow local deflections to take place, but this is not usually advisable as secondary stresses are imposed on the lining.

Bearing Performance: Progress in the development of machinery which involves a multitude of factors can only be directed by the study of failure either as experienced in the field or as produced intentionally by test to destruction. By far the majority of bearing failures are, in the authors' experience, due to faulty assembly.

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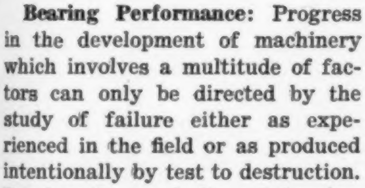


Fig. 2—Schematic interpretation of conditions producing trapped volume lubrication

Design Abstracts

seizure, excessive wear and mechanical breakdown.

Seizure

Bearing seizure can only occur when the hydrodynamic film breaks down, but the breakdown of such a film does not necessarily result in seizure of the whole bearing. This is to a large extent dependent on the properties of the metals and lubricant involved. From the point of view of lubricated bearings, the following conclusions are of particular importance.

Friction results from adhesion taking place between localized high spots. The "welds" so formed may be stronger than either of the metals, so that rupture takes place within one or other of the metals, resulting in transfer of material from one surface to the other. Such welding and transfer of material is reduced, but not avoided, by the presence of a boundary film of lubricant.

When welding takes place at high points, the temperature at these points may rise to the melting point of the softer material.

Oxide films on the metals exercise a profound effect in reducing adhesion and soft, strongly-bonded, and rapidly-forming oxide films are desirable.

Combinations of metals and lubricants which form a soap which is solid at the operating temperature are particularly effective in reducing friction.

Friction is reduced by the presence of a very thin film of a soft material on a harder material.

Occurrence of adhesion is not universally accepted as the primary factor in friction, and a recent paper puts forward evidence that mechanical interlocking of surfaces, roughened by plastic deformation, is the primary factor. The weight of evidence appears, however, to be in favor of the simple adhesion theory.

These conclusions lead to the following conditions in the choice of a bearing material:

1. There should be as large difference between the shear strength of the material of the two opposing surfaces as possible. When welds

form at high spots, a large difference in hardness will tend to cause welds to fail consistently in the weaker material, thereby confining damage to one surface. Damage is not then self propagating. For similar reasons, there should be a considerable difference between the melting points of the opposing surfaces.

2. Bearing materials should preferably form oxide films which are tenuous, strongly bonded, and not excessively hard.

3. Bearing material and lubricant should react to form a soap, preferably one which is solid at operating temperatures.

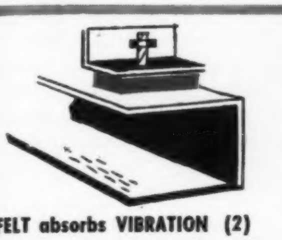
Practical experience tends to support these conclusions, for the metals found satisfactory in practice possess most of the desirable qualities. Tin, lead, cadmium and their alloys have low shear strengths and melting points and readily react to form soap. Tin oxide is tenuous, though hard, while lead and cadmium oxides are soft, though not tenuous. Copper has appropriate oxides, though its melting point and shear strength are rather high. Aluminum has an unsuitable oxide but is otherwise intermediate in its properties.

Hydrodynamic Film

The foregoing factors are of importance when the hydrodynamic film breaks down, that is, in the region C of the ZN/P curve shown in *Fig.* 1. There is also considerable evidence that the material of the surfaces affects the stability of the hydrodynamic film. It has been shown that the minimum of the ZN/P curve occurs at a lower value in the case of babbitt than in the case of copper-lead, indicating that the hydrodynamic film was more easily maintained with babbitt than copper-lead, other conditions constant. One of the present authors has put forward evidence to show that partial hydrodynamic lubrication commences to operate at lower speeds with tin and lead-base alloys than with copper-lead, cadmium-nickel alloy being intermediate. These findings were based on experiments conducted with hemispherically-ended sliders, sliding on

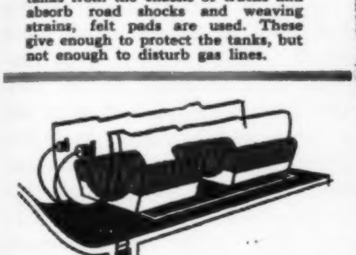
FELT for GASKETS (1)

There has to be a seal between the lid and body of electric power transformers, to exclude dirt and moisture, hold in the oil that insulates and cools. Felt is widely used for this purpose.



FELT absorbs VIBRATION (2)

To isolate spring-suspended gasoline tanks from the chassis of trucks and absorb road shocks and weaving strains, felt pads are used. These give enough to protect the tanks, but not enough to disturb gas lines.

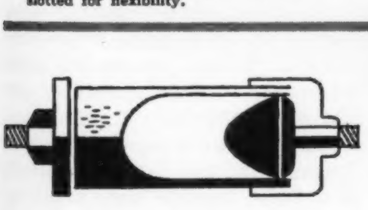


FELT for LUBRICATION (3)

Many locomotive journals are lubricated by felt, cut to fit the hub. Pumped oil is fed evenly, waste-grabbing eliminated. Length of life of the felt is estimated at 75,000 to 100,000 miles of perfect lubrication.

FELT for POLISHING (4)

The final polishing operation on steel cutlery changes the finish from dull to bright. About .002" of steel is removed by felt polishing discs, fed with fine pumice. The discs are slotted for flexibility.



FELT in REFRIGERATION (5)

In refrigerators and air conditioners there is usually a cartridge filled with a dehydrating agent. Felt is used as a filter to prevent small particles of the drying crystals being carried into the refrigerant line.

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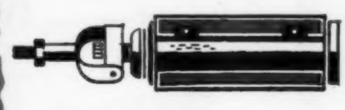
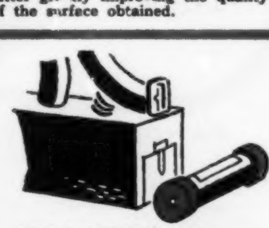
FELT CLUTCH FACING (6)

Industrial sewing machines have a clutch, which must take hold quickly, yet smoothly and firmly. These clutches operate properly when faced with felt discs cut to size and shape. This is a frictional application of felt.



FELT for HONING (7)

The cylinders of internal combustion engines are finished by honing. The honing head carries abrasive stones alternated with strips of felt, the latter greatly improving the quality of the surface obtained.



FELT for SHOCKS (8)

Pneumatic carriers of both large and small sizes are in wide use. American supplies special felts for them. One type is used as a bumper head on the carrier to absorb shock on delivery; another is an air-pressure seal.

FELT for FIRE EXTINGUISHERS (9)

Hand-pumped fire extinguishers use felt washers for lubrication, for holding compression, as a bumper for the upstroke, and as a cushion for the nozzle. American also supplies flame-proofed felt for airplanes, theatres, etc.

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a flat steel plate. Experiments were made with plates covered with extremely thin lubricant films and with excess of lubricant, and the difference attributed to fluid lubrication. It was further shown that the extent of fluid lubrication, so determined, is dependent on the hardness of the alloy. Maximum degree of fluid lubrication will be obtained when the surfaces can adjust themselves to a suitable hydrodynamic configuration, and materials which are readily capable of plastic flow will adapt themselves more readily than materials which are not.

Bearing Wear

Excessive wear results from the breakdown of hydrodynamic films and the consequent boundary friction. The previous factors considered again apply. Such wear will be minimized by using materials which favor fluid lubrication and which have inherently good frictional properties under boundary conditions. Wear which results from incompatibility of surfaces is likely to be rapid and "catastrophic," that is, occurring at an increasing rate as it proceeds. It appears, however, that with reasonably good materials in a well-designed bearing, the wear due to boundary friction is negligible.

Roach examined in considerable detail the effects of an abrasive-containing lubricant. Using an Underwood bearing testing machine, Roach investigated the effects on wear and bearing temperature of adding an abrasive of known particle size. The curve shown by Roach, in which wear is plotted against particular size, goes through the origin, implying zero wear with zero particle size. If, however, Roach's results are plotted as rate of wear against square root of particle size, a closer approximation to a straight line is obtained, and this cuts the abscissa at a finite particle size, implying that no wear is experienced until the abrasive particle exceeds a certain size.

Roach goes on to compare the behavior of various bearing materials in the presence of a lubricant

containing a known amount of crystalline alumina, of a nominal particle size of 28 microns, using temperature rise as a criterion. On this basis, bearings with a surface layer of at least 0.0001-inch of a babbitt alloy are very much better than copper-lead or aluminum-alloy surfaces.

Mechanical Breakdown: To avoid seizure and to reduce wear to a minimum, the softer, low-melting point metals and alloys are, both in theory and in practice, to be preferred. A bearing must, however, sustain a definite load imposed on it through the oil film and the intensity of the load which bearings are required to sustain is continuously increasing as engine development proceeds. Constant loads can generally be met with little difficulty and it is only in some specialized applications, such as rolling-mill bearings, that problems of failure by plastic flow are encountered. Engine loadings are, however, fluctuating in character and tend to lead to failure by fatigue. The "crazed" appearance of a bearing which has failed by fatigue is all too familiar to bearing users and manufacturers.

Thus, there is a fundamental incompatibility between the properties required in a bearing material; low melting point, softness and low shear strength on the one hand and high fatigue strength on the other. The problem of the bearing designer and the metallurgist is to find the best possible compromise for particular circumstances. One line of approach which is almost universally used in high-duty engine bearings is to use a duplex material having a soft lining rigidly bonded to a stronger backing. It has been known for some time that, in general, the thinner the lining, the greater the resistance to fatigue failure.

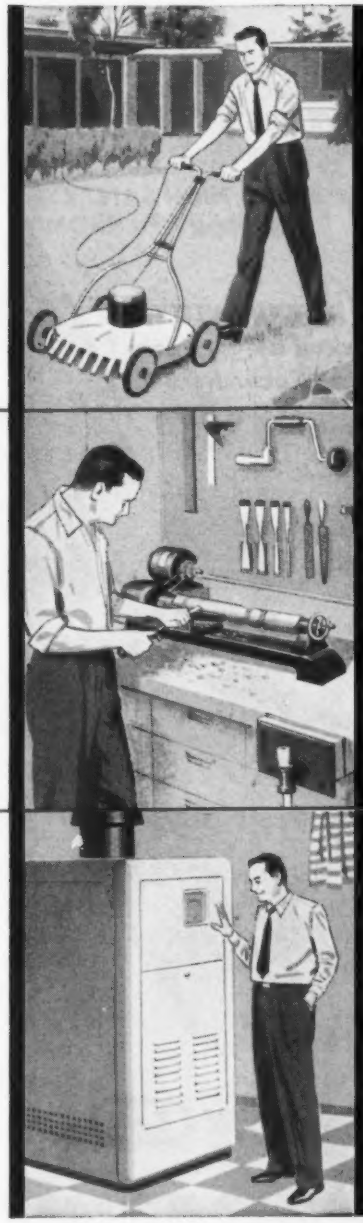
Schaefer has published curves showing the relation between lining thickness and fatigue life. These curves indicate that fatigue life decreases as lining thickness is increased up to about 0.015-inch; increase above 0.015-inch has little effect. It should be noted that these values relate to a particular

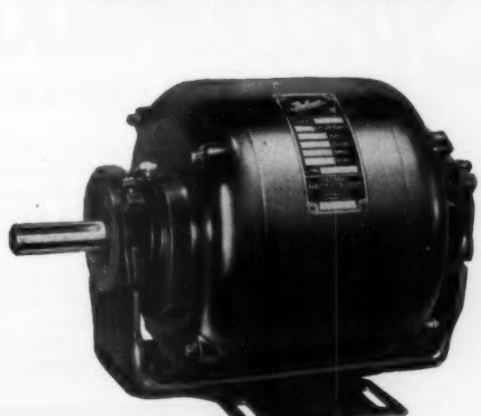
design of bearing, and do not necessarily apply quantitatively to other designs.

Practical Implications: The lines along which bearing materials with favorable combinations of properties can be sought are fairly well-known. Lining material should be fairly soft at operating temperature to assist fluid film formation and enable some abrasive in the lubricant to be tolerated. It should have a low-melting point, a suitable oxide film and be capable of forming a soap with normal lubricants. Elastic modulus should be low. The lining should be used in the form of a layer rigidly bonded to a backing of high modulus. The layer must be thick enough to provide adequate tolerance for abrasive and yet thin enough to have adequate fatigue strength. In practice, two other lower limits are imposed on the thickness of the lining. First, the lining must be sufficiently thick to take up the full extent of any wear which will take place. Secondly, the lining should be capable of some degree of plastic flow, in order that, where necessary, some degree of edge loading can be accommodated. It is largely on account of these factors that trimetal bearings have been developed, consisting, for example, of a steel backing, a 0.015-inch interlayer of copper-lead, and a 0.0015-inch overlay of lead-tin alloy. The limiting fatigue loading of such a bearing is, at least theoretically, less than that of a bimetal bearing consisting of 0.0015-inch of lead-tin on steel. The trimetal bearing has the advantages of having a greater capacity for adapting itself to edge loading (by plastic flow of the interlayer) and of possessing tolerable bearing properties if the bearing surface proper should wear through.

Thus, qualitative information is fairly complete in that most of the factors involved and the direction in which they operate are known. Quantitative information is still lacking.

From a paper entitled "Functions of Materials in Bearing Operation" presented at a general meeting of the Automobile Division of the In-





stitution of Mechanical Engineers in London, England, December, 1953.

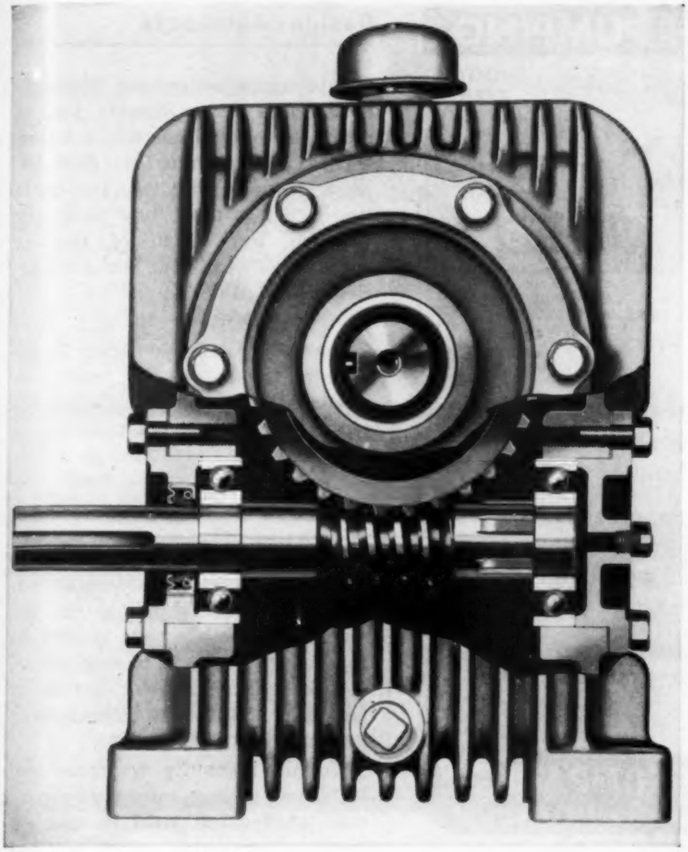
Creative Design—Can It Be Taught?

By George B. Du Bois
Mechanical Engineering Dept.
Cornell University
Ithaca, N. Y.

CREATIVE design or creative engineering have become popular words in recent advertising, and there seems to be a growing desire at several engineering colleges to offer courses in creative design. Advertising may have added glamour to the title, but it is undeniable that the first assembly of any new design serves to create something new, and the design doubtless required considerable ingenuity. The question may be asked, "Can ingenuity be taught or is it something a man is born with?" Actually ingenuity is a combination of several factors, only a few of which are inheritable. By making a creative design course an elective one, total attention can be devoted toward improving ingenuity. Actually this field is so nearly untouched that rapid initial gains easily can be made. As in most other skills, the initial practice produces a large improvement. There is no problem with lack of ingenuity in our students; on the contrary their work has sometimes been amazing. People have a sort of fear of doing anything they have never done before, and this mental block often can be overcome easily by practice resulting in assurance.

Creative Ingenuity: The approach to ingenuity is a field almost untouched; it is like a course in geometry would be with no textbook of accepted theorems. A number of general principles seem to be recognizable, but more widespread effort is needed to recognize principles and collect examples illustrating them.

In teaching creative design, each principle or suggestion should be illustrated by real examples. We



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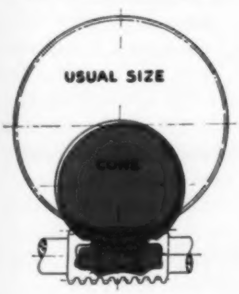

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try to avoid suggesting improvements on a design directly, but instead use suggestions which follow to aid the individual to find improvements of his own. The question is not so much how to do the problem, but which of two or three ways works out best for particular design needs.

Ingenuity and Memory: If we attempt to analyze the process of ingenuity in design, we find it often consists briefly of thinking of some old ideas arranged in a new way to suit a current need. The process is similar to the memory process, and a good memory for construction methods is useful. There has been considerable advance in understanding of the memory process which is said to function by the laws of association. It is known to be easier to remember ideas if they are in associated groups.

To aid ingenuity we must collect surrounding information and associated detail until we can remember something which fits the requirements. The problem is to reduce the mental step to a size the brain can see across. The following suggestions can only lead us up to this point.

Changes in Design: Few people realize that there is a time when improvement changes are welcome, and should even be sought. This time is before the production tooling has been made, and preferably while the layout design is in progress. The realization is growing that more time, and more thorough work on design development, can reduce the number and cost of changes required later.

Types of Design Problems: The problem of obtaining a new idea can be broken down into two categories. There are those which seek an improvement for a known or unsatisfactory method and those where a first solution is unknown. The suggestions for the first type can be called transition methods. The second type can be called initial solution methods. The following nine examples serve to show how the transition method type of

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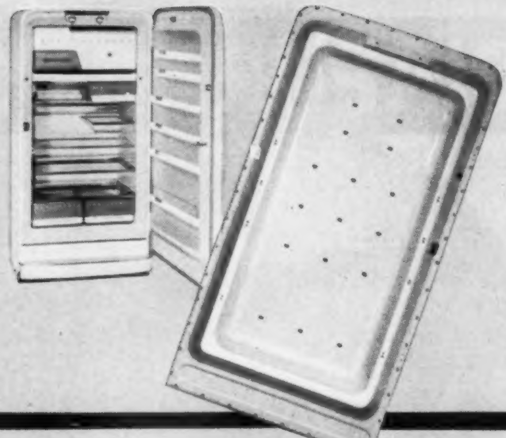

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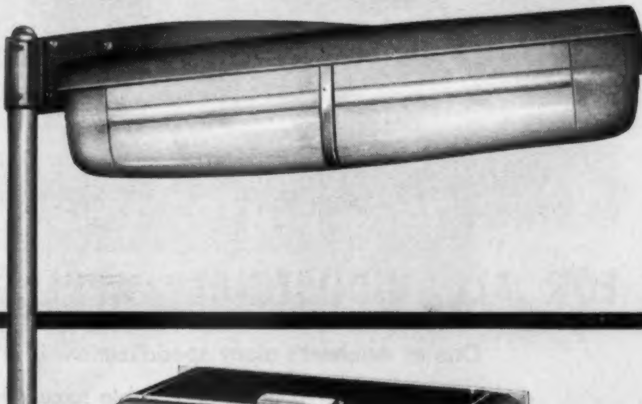

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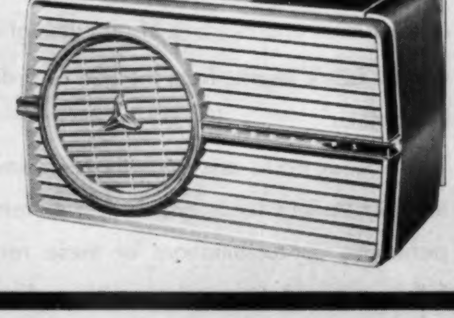

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Transition Methods

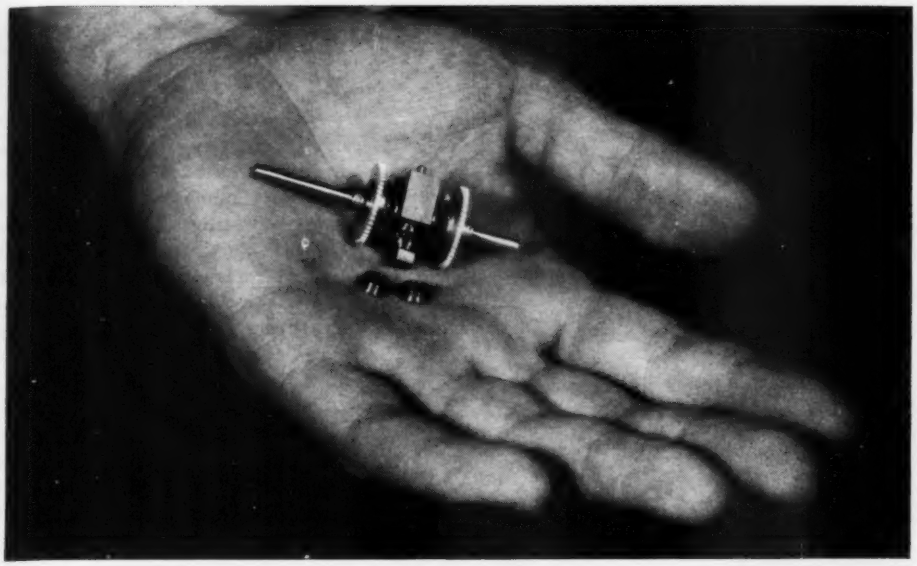
BUILD ENTHUSIASM WITH THOROUGH SELLING: One of the chief aids that can be given a designer's ingenuity is to make sure he is sold thoroughly on the advantages of successful completion of a problem. In industry especially, supervisors will find that the time required to sell the problem will be saved later on. Make sure all the weak points of a new idea are discussed, for if a man has any mental misgivings it in some way hinders the required mental processes.

DESIGN FOR A PARTICULAR METHOD OF MANUFACTURE: Modifications in existing designs easily are made to supply variations using cast, welded, sheet-metal, or molded-plastic designs for the same purpose. Combined units and subassemblies are useful. Consider a smooth or rounded appearance of the exterior.

MINIMIZE DISADVANTAGES: Some successful designs are the result of a thorough effort to minimize a difficulty almost out of existence. Hydraulic brakes are in almost universal use in motor vehicles in spite of the evident safety problem in regard to leakage, which might have discouraged the original designer.

USE CLASSIFICATION METHOD: In the early days of controllable-pitch propellers for aircraft, at least two different mechanical designs were later superseded by the well-known electric and hydraulic types now in common use. It would have been well at that time to consider the names of other classifications of designs such as these. There are vehicle brakes in almost every classification, and the names of types of brakes can be used to identify the names of various classifications, such as air brakes, electric brakes, etc. A designer on a new problem should certainly consider mechanical, electrical, hydraulic, and pneumatic solutions or combinations of them.

CONSIDER PHYSICAL INVERSION: It takes only a few years of experience to find several cases in



Design Abstracts

which the opposite way turned out to be better. A mechanical engineer easily should recognize two or more sequences, three principle axes, and numerous inversions of levers, gearing, four-link chains and planetary gearing on a basis of relative motion. It is a good rule to always consider the other way around a bush—if you don't, your competitor may show you how.

CONSIDER MENTAL INVERSIONS: In addition to physical inversions, try inversions of words or thoughts, for example, a turbine rotor mounted on a sleeve around a stationary shaft; versus a turbine rotor mounted on a shaft turning in a stationary sleeve. A little practice with quiz problems will develop a strong realization of the possibilities of inversion of words or thoughts.

USE TEAMWORK: Talk your problems over with other people. It is a proverb that it is easier to do the other fellows job than to do your own. People naturally seem to take opposite points of view, and by human nature attempt to beat each other to an idea. Conversations breed ideas.

NEVER SAY IT CAN'T BE DONE: An inversion of this rule works even better. If you feel like saying can't, invite yourself to do so, but write down a list of qualifying limitations. Any possible solution must lie in some qualifying phrase. When a beginner is about ready to say it can't be done, it is probable that he really has discovered what he wants to do if he could; the answer often lies in another view.

Initial Solution Methods

Occasionally a problem arises for which even the first solution is not apparent. A number of suggestions can be enumerated briefly:

DON'T DISCARD AN UNSATISFACTORY SOLUTION TOO SOON: A little work on it, setting up detail sketches, may reveal a solution or a method of minimizing the trouble. Give your ingenuity a chance.

DIVIDE A PROBLEM INTO STEPS: When the solution is explained it will probably consist of three or

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Bundyweld nickel and Monel tubing are sold by distributors of nickel and nickel alloys in principal cities.

more steps. The problem is to recognize the steps beforehand. Make sketches or blank diagrams of each step. Naming the steps may help. They will be named later if the solution is put to use. As an example, in type-setting machines it was a milestone to recognize the steps of (a) composing a line, (b) adjusting its length and (c) redistributing the type after use.

APPLY THE BLIND MAN METHOD: Consider hand methods in detail, converting to equivalent mechanical steps.

CONSIDER MACHINE METHODS: Evaluate rotation, reciprocation and power.

USE THE "X" METHOD: Work out the other phases leaving one step unknown for further consideration. Sketch a space for step X and all that is known about it. This gives more detail for the unknown step so that diagrams of the required motions can often be set up.

CONSIDER THE PHYSICAL, CHEMICAL, OR ELECTRICAL LAWS OF NATURE: Some known effect might be useful. For example, will it be gravity feed, or mechanical, air pressure, etc.

Results

For utilization in a classroom, this type of creative approach eases the problem of criticism of first efforts, since suggestions are welcomed to aid in obtaining another solution. After a few weeks of experience including reviews of what others did, the students will readily agree that:

1. The first solution is often not the best.
2. Working on the second and third trials increases the "know-how" needed to cope with a problem.
3. Best ideas come after sketches have progressed in detail.
4. Never say can't without qualifying it to aid you to find the answer.
5. Evaluation is more effectively done by comparison.

From a paper entitled "Instruction in Creative Mechanical Design" presented at the Annual Meeting of ASME in New York, N. Y., December, 1953.

New Machines

Materials Handling

Conveyors: Vibrating units convey and/or screen and distribute a variety of bulk materials. Are compulsory driven, swinging-mass type in which two troughs or screens, mounted one above the other, are connected by leaf springs. The whole assembly, supported on guide links, follows the movement of the eccentric drive. Model 2-B, with a maximum capacity of 80 tons per hour, is available in sections up to 65 ft long with a 20-in. trough and in sections up to 30 ft long with 40-in. trough. Model 6-F, with maximum capacity of 280 tons per hour, has sections up to 175 ft with 20-in. trough and 100 ft with 40-in. trough. Sections can be set in to any desired length. *Syntron Co., Homer City, Pa.*

Fork Truck: Gasoline powered Yardloader, for outside and rough-surface work, has 4000-lb capacity and travel speed of 14 mph. Truck has low-pressure hydraulic system; motor driven gear pump supplies oil under pressure to either lifting or tilting cylinders. Pump speed is controlled by engine acceleration, excess oil being by-passed to the reservoir. Overall size is 100½ in. long, excluding forks, and 50 in. wide; ground clearance is 5 in.; turning clearance, 92 in. *Baker-Lull Corp., Minneapolis, Minn.*

Pneumatic Unloader: Vacu-Veyor has capacity of 40,000 lb per hour and moves bulk materials up to 150 ft vertically and 200 ft horizontally. Flexible hoses permit travel around corners and other obstacles. Used for any dry, bulk powders, grains, pellets or crystals, unit is portable and can be powered with electricity or a gasoline or diesel engine. *Vacu-Blast Co. Inc., Belmont, Calif.*

Unwind Stand: Model UA single-position unwind stand incorporates an air-actuated brake made by Linderman Devices to regulate tension of unwinding webs such as

paper, board, cellophane, foil or film. Brake can be controlled either through a manually operated pressure regulator or a pressure modulating device for automatic adjustment of braking efforts as the web unwinds. Operating on ordinary factory air line pressure, the unwinder has quick-acting split-latch type sleeve bearings and running adjustments for side lay and squaring. Sizes are available for various roll diameters and widths. *Dilts Machine Works Div., Black-Clawson Co., Fulton, N. Y.*

Metalworking

Tube and Bar Straightener: Model 4 KTC seven-roll straightener employs cluster roll arrangement which confines work to pass line without guides. Straightens tubes ranging from 3⅝ to 12 in. OD and bars from 3⅝ to 7½ in. diameter. Clusters consisting of a large driven roll with two opposed idler rolls are located at both entry and delivery ends of the machine. Between the clusters is an unopposed pressure roll. Power-operated screwdowns and automatic co-ordinated angling of rolls permit complete changeover from minimum to maximum size in less than three minutes. Roll angles are automatically adjusted to the proper setting for size of material being straightened, with full contact between work and rolls. *Sutton Engineering Co., Bellefonte, Pa.*

Milling-Centering Machine: No. O Centrmil handles work from ¾ to 3½-in. diameter and 5 to 48 in. long. Simple adjustment facilitates miscellaneous shaft work. After shaft has been located and clamped, rocker type milling head mills one end. Milled end is then manually centered with a ¾-in. center drill. Part is reversed and located from center or end previously machined for milling and centering opposite end. Power feed to center drill and power clamping can be provided. *Sundstrand Machine Tool Co., Rockford, Ill.*

Impact Punch: Redesigned Series 4 solenoid-operated Electropunch delivers variable impacts up to 3500 lb at rates of more than 125 blows per minute. Base has in-

YOU can't expect the engine in your product to deliver maximum power to the wheels — unless the drive line is specifically suited to the operating characteristics of your product.

MECHANICS Roller Bearing UNIVERSAL JOINTS and PROPELLER SHAFTS are factory, field and road tested for torque, overload, shock, angle, alignment, balance, speed changes, reversals and stamina.

Regardless of the type and size of your product, MECHANICS makes joints and propeller shafts that will meet its power transmission needs exactly.

Whatever your drive line problem may be, MECHANICS engineers are prepared with the experience and facilities to help solve it.

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Borg-Warner • 2032 Harrison Ave., Rockford, Ill.

creased working surface and a skirt protects recessed hand switch. Strengthened adjustable bracket affords better support and alignment for solenoid which is enclosed in a perforated steel housing with a removable cover. Adjustable, cushioned return stop on top of solenoid limits return stroke and reduces noise and wear. Operable by hand, foot or automatic switch, punch is used for staking, riveting, marking, cutting, etc. *Black & Webster Inc., Newton, Mass.*

Small Rolling Mills: Line of mills for flat work and for producing wire and other shapes; combination mills for both flat and wire rolling; and combination flat and wire mills with bull block attachment for wire drawing can be made to meet individual requirements through the use of interchangeable standardized parts. The 100 series 3-in. mills and the 200 series 4-in. mills are both capable of heavy reductions to close tolerances. The 3-in. mills have a 5-in. roll face width; because of increased roll diameter and stiffness, 4-in. mills are made 6 in. wide. *Stanat Mfg. Co., Long Island City, N. Y.*

Cold Crimping Machine: Uses strip terminals in continuous form. Automatically attaches, crimps and cuts off in one operation. Interchangeable rolling tools are set up in an accessible, removable die set which can be placed elsewhere for faster tool changing under favorable lighting conditions. Extra die sets are available for keeping tools set up ready for immediate placement in machine. Operation is by air pressure. *Patton-MacGuyer Co., Providence, R. I.*

Manifold Processor: Special Transfer-matic machine for drilling, boring and milling intake manifolds processes parts at the rate of 218 per hour. Seven stations including loading and unloading perform 22 drilling, eight chamfering, two reaming, 12 tapping, eight boring, and two milling operations. Two parts are machined at one time at each station. Palletized work holding fixtures carry the manifolds automatically from station to station. Power wrenches are used for operating the work

New Machines

holding fixtures automatically. Machine also has preset tools, built-in chip conveyor and automatic cleaning unit for removing chips from fixtures. *Cross Co., Detroit, Mich.*

Power Presses: Electromatic power press and Multi-power press are available with 13-ton capacity. Electromatic press, Model 56-E, has electrically operated nine-point jaw clutch. It can be adjusted without stopping for single stroke nonrepeat or continuous action. Model 56-J is equipped with Multi-power nine-point mechanical type jaw clutch. Both presses have one-piece heat-treated alloy crankshafts, split bronze-bushed main bearings and large ram area. *Famco Machine Co., Kenosha, Wis.*

Office Equipment

Adding Machine: Redesigned portable electric model has "multiflex control" which replaces the repeat key with a multiply bar located at the right of the numeral keys. Long enough to be used with either the motor bar or subtract key, the multiply bar is held down with the motor bar for direct multiplication. Short-cut multiplication is performed by holding down the multiply bar and subtract key. Machine cycles at 150 rpm and is sound conditioned to a nonirritating pitch. Performs adding, subtracting, nonadding, subtotaling, totaling, simplified multiplication and printing of credit balances. Changing from one type of calculation to another is accomplished by shifting keyboard entry. *Underwood Corp., New York, N. Y.*

Dictating Machine: Dupli-Voice dictating and transcribing machine employs a magnetic belt to record. Belt reproduces with the same fidelity as tape or wire and can be reused many times. Machine measures $5\frac{1}{2} \times 10$ in. and weighs 12 lb. *Dupli-Voice Co., Inc., Algonquin, Ill.*

Typewriters: Made in Sweden, Halda Star standard typewriter has a magnesium carriage which operates with ball bearings. Margin setting is accomplished by depressing a key for setting, resetting or clearing margins. Five per-

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SAVES YOU MONEY 10 WAYS:

- Boosts output
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- Reduces "down time"
- Simplifies machine design
- Reduces operator fatigue
- Cuts changeover time
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Get the Facts! Ask for Bulletin D-2102. It describes and illustrates features, applications, components and operation; dimensions and characteristics also are included.

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New Machines

manent tabulation settings are available for statistical typing. Plastic keys are shaped to fit fingertips. Machine is finished in light green. Portable model, with all-steel body, is housed in a tan, luggage type steel case, weighs 12 lb. *Facit Inc., New York, N. Y.*

Copying Machine: For small volume work, Bambino model is slightly larger than an electric typewriter. It copies anything that is written, typed, printed or drawn on a sheet of translucent paper up to 9 in. wide, any length. First copy is made in seconds; up to 200 copies can be made in an hour. *Ozalid Div., General Aniline & Film Corp., Johnson City, N. Y.*

Portable Tools

Heavy-Duty Hand Grinder: Series 35 hand grinder has 1/4-hp continuous duty motor which delivers speeds up to 35,000 rpm and can be used with tungsten carbide mills, high-speed cutters, mounted wheels and points, abrasive bands, etc., for grinding, milling, finishing, and polishing materials ranging from wood to hard alloy steel. Overall length, including chuck, 10 3/4 in. Body diameter is 2 7/16 in.; small diameter, 1 5/16 in. Weight, approximately 3 lb. Available for 115 or 230 v dc or ac, 0 to 60 cycle operation with 1/8 or 1/4-in. collet chuck. *Dumore Co., Racine, Wis.*

Portable Spot Welder: Has enclosed electronic timing control with timing cycle ranging from 1/60 to a full second. Welds most aluminum alloy sheets and mild and stainless steel including galvanized and cadmium plated. Output exceeds 10 kva. Welder has pistol grip and is also available with a stand for use as a foot operated floor model. Unit weighs 26 lb. *Ampower Products Co., Oak Lawn, Ill.*

Lathe: Portable, precision lathe for metal, wood and plastics measures 10 1/2 in. long, 3 3/4 in. wide, 7 1/4 in. high and weighs less than 12 lb. Takes work up to 4 in. in length between centers, has 3-in. swing and 1/8-in. collet capacity. Has 1 1/4-in. driver plates, two lathe centers, tool post, spacer and rocker, adjustable tailstock, tool

MAXITORQ
Overload Release
CLUTCH

minimizes machine Down-Time

AT THE
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The Maxitorq automatic Overload Release Clutch was designed especially to protect high-speed machinery, and we are gratified to have testimonials such as the following:

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"Over a period of a year, wrapping millions of blades per week, Maxitorq Clutches have eliminated machine down-time except in a few minor instances. Thus, more constant production has been maintained in the packaging department."

When an accidental overload occurs, the clutch automatically releases, stopping the machine, preventing damage to machine and product. When the jammed condition has been cleared, the clutch is re-engaged and the machine is again in operation. By means of a simple finger-tip adjustment, the clutch is set to transmit the normal running load.

There are six sizes, ¼ to 5 h.p. @ 100 r.p.m.; max. working torque ft. lbs. 13 to 263. Maxitorq "floating" discs prevent heating in neutral, and all assembly, take-apart, and adjustments are manual; disengagement is instant and complete. Submit your clutch problems to our engineers for practical solutions.

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Good engineering designs depend on low cost.

SIMPLE DESIGN CHANGE TO STEEL CUTS COST FROM $1.15 TO 31¢

BEFORE any product design is accepted, the manufacturer asks, "Can it be built for less money?" Unless your designs pass this test they are likely to be rejected.

Knowing how to use welded steel gives you the advantage in developing any product for lowest cost manufacture. That's because steel is three times stronger than gray iron, two and one half times as rigid, and costs only a third as much per pound. Therefore, where stiffness or rigidity is a factor in a design, less than half the material is necessary.

Here, for example, is how one resourceful engineer put these qualities to work:

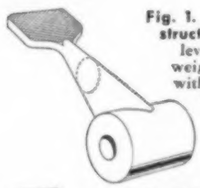

Fig. 1. Traditional Construction. Machine foot-lever, 10 inches long, weighs 6 pounds. Cost with broached keyway is $1.15.

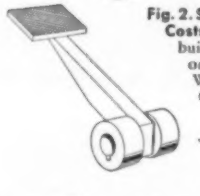

Fig. 2. Simple Steel Design Costs 41% Less. Can be built by the shop with only saw and shears. Weighs 2.7 pounds. Costs 68¢ complete with keyway.

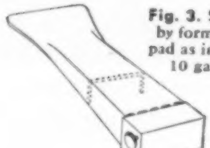

Fig. 3. Saves 53% Cost by forming lever arm and pad as integral piece from 10 gauge metal. Weighs 2.5 pounds. Costs 54¢.

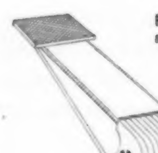

Fig. 4. Saves 73%, Eliminates Broaching. Hub with integral key is produced by stacking stampings in assembly. Arm is 10 gauge, brake formed and welded to hub. Cost is only 31¢. Weighs 2.2 pounds.

HOW TO DESIGN IN STEEL AND CUT COSTS

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New Machines

bit, lathe dog, test rod and Allen wrench. *MasterSon Engineering Co., Los Angeles, Calif.*

Processing

Vacuum Metallizer: Vacuum coating unit type LC1-30, with normal production rate of four cycles per hour, has 30-in. diameter horizontal chamber with fixtures and filament holder. Chamber is mounted on a steel cabinet which contains filament power supply, fixture rotating motor and a door-mounted electric control panel. Pumping system, including a high-speed 14-in. oil diffusion pump, a small mechanical holding pump and a 100-cfm mechanical pump for roughing and backing, is located at the rear of the chamber and is completely valved. Unit may be supplied with a stationary fixture for single surface coating or with two rotating fixtures, each holding 12 riser rods. *Consolidated Vacuum Corp., Rochester, N. Y.*

Coating Machine: For coating coiled stock. Machine automatically feeds material from uncoiler, deburrs edges, coats both sides with drawing compound and feeds the press. Four sets of rolls are employed: hardened smooth ground steel for deburring, rubber covered nip rolls which feed coiled steel from uncoiler, doctor rolls of smooth ground chrome plated steel, and smooth ground rubber-covered coating rolls. Circulating pumping unit with 5-gal sump tank located under rolls supplies drawing compound to doctor and coating rolls. Calibrated handwheels provide adjustment of doctor rolls in and out against coating rolls to permit control of thickness of application. Machines are built to handle stock widths to 60 in. in thicknesses from 0.005-in. *Union Tool Corp., Warsaw, Ind.*

Testing and Inspection

Ring Gage Tester: Bench type electronic precision measuring instrument gages lead, taper, straightness and concentricity of straight and tapered plain and thread ring gages up to 8 in. ID and 10 in. OD. Instrument consists

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DOW CORNING

Silicone News

No. 5 of a Series • PUBLISHED BY DOW CORNING CORPORATION, MIDLAND, MICHIGAN

New Data Reveals Effect of Heat Aging on Properties of High Temperature Laminates

Comparative studies of the effect of heat aging on light weight structural materials indicate that the better resin bonded glass laminates compare very favorably with light metals at high temperatures. These studies made recently by our engineers and summarized below in Table I show that the laminates have greater tensile strength than the metals at 500 F after aging for as little as half an hour at that temperature.

Tensile strength to weight ratio for the laminates is also superior to that of the metals under the same conditions. Measured at 500 F after 200 hours at that temperature, the tensile strength to weight ratio times 10^{-2} for the silicone-glass laminates is in the range of 4 to 5 times comparable values for the light metals.

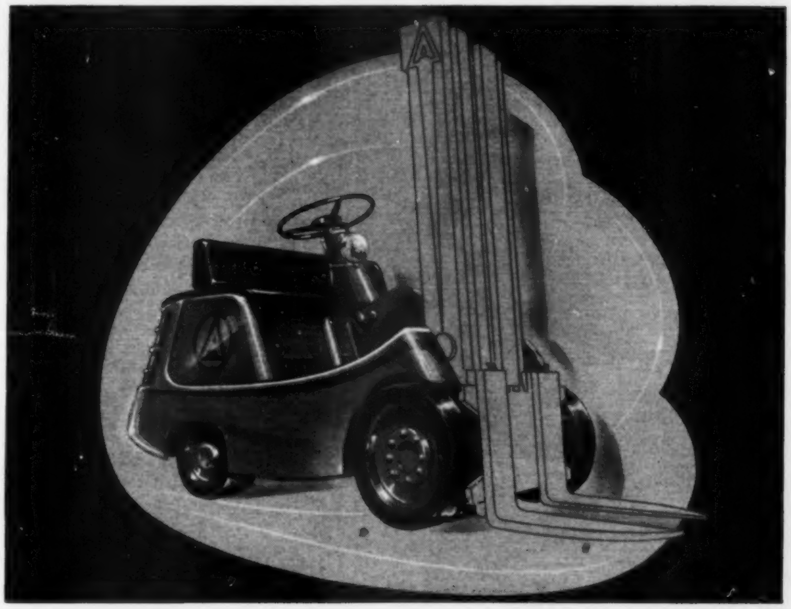
The organic resin bonded glass laminates show excellent short term retention of mechanical strength at 500 F. Strength falls off quite rapidly, however, with increasing time. The mechanical properties of the silicone-glass laminates at 500 F decrease during the first half hour and then improve or remain unchanged with increasing time. Weight loss, water absorption and dielectric properties of the silicone-glass laminates are markedly superior.

Included in this study were glass laminates bonded with several new, heat resistant low pressure phenolic, polyester and

(*continued pg. 2*)

TABLE I
EFFECT OF HEAT AGING ON HIGH TEMPERATURE LAMINATES AND LIGHT METALS

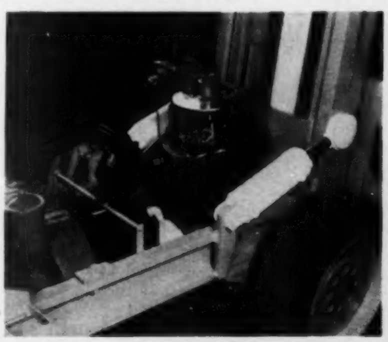
Property	High Pressure Silicone	Low Pressure Silicone	Low Pressure Polyester	Low Pressure Phenolic	Aluminum	Magnesium
Flexural Strength, psi						
As received	45-47,000	27-32,000	45-55,000	44-66,000		
At 500 F after						
½ hr at 500 F	5,000	12-15,000	35,000	41-50,000		
200 hr at 500 F	19,000	12-15,000	11-25,000	9,000		
Tensile Strength, psi						
As received	26-32,000	35,000	33-36,000	22-27,000	50-65,000	39-42,000
At 500 F after						
½ hr at 500 F	8-10,000	22-25,000	30-33,000	24-29,000	7-13,000	8,000
200 hr at 500 F	18-20,000	20-22,000	9,200	11-15,000	5-9,500	6,000
Density, g/cc	1.65-1.80	1.70-1.85	1.65-1.85	1.50-1.80	2.7	1.76-1.80
Tensile Strength to Weight ratio x 10^{-2}						
½ hr at 500 F	5.3	13.6	18.0	16.1	3.7	3.5
200 hr at 500 F	11.0	12.2	5.3	7.9	2.7	3.3
Weight Loss, percent 200 hr at 500 F	0.35-1.31	0.1-0.2	9.00	11.2		
Water Absorption, percent (After aging 200 hr at 500 F)	0.21	1.0-1.3	2.00	9.00		
Dielectric Strength, v/m, ⅛" electrodes						
As molded	80-100	300	90	200		
Aged 200 hr at 500 F	80-100	80-100	70	60		
Power Factor, percent As received, at 10^5 cps	0.14	0.22	1.2	2.7		
Dielectric Constant As received, at 10^5 cps	3.90	4.13	4.00	4.31		
Arc Resistance, sec. As received	300	350	185	165		



Designers Use Class H Insulation to Sell More Electric Machines in Buyer's Market

Instead of viewing with alarm our entry into a more normal competitive economy, Automatic Transportation Company has prepared for it. They have developed a new product with many strong selling features, including more continuous operation with minimum maintenance cost.

"Dynamotive", Automatic's newest entry in the materials handling equipment market, has so many new features that it bears little resemblance to the lift truck of 5 or 10 years ago. The first gas-powered, electrically driven fork lift truck, it has an electrical system that assures long service life by making it impossible to reverse the direction of the truck while it is in motion.



Engine wear and fuel consumption are minimized by an electrical transmission with its inherent overdrive feature that

(*continued pg. 2*)

FOR DATA RELATING TO THESE ARTICLES, CIRCLE REFERENCE NUMBER IN COUPON ON NEXT PAGE

MORE

of a 32 × 12-in. cast-iron base on which is mounted a rotating face plate, an electronic pick-up stylus on a gaging arm which adjusts vertically on a ball-bearing slide mounted column, a micrometer screw for use with precision end measures, and an electronic amplifying indicating unit. Face plate can be swung through a 90-degree arc to rest upon a horizontal precision sine bar fixture. Ring is centered in relation to circular lines scribed on the plate and locked in position with three clamps which are adjustable for rings of various thicknesses. Electronic amplifier with an amplification of 2000 to 1 can be remotely located for operator convenience. Interchangeable ball point gaging tips in sizes 0.030, 0.050, 0.080 and 0.125 are supplied to permit gaging of most threads. *Sheffield Corp., Dayton, O.*

Torque Tester: For precision ball bearings; compares breakaway, starting and running torque and evaluates race roughness and dirt in bearings. One control enables operator to raise the voltage on an eddy current motor until it just starts the bearing, at which point a meter reading gives a comparative measure of starting torque, while at reduced voltage a steady state running torque is shown. On another meter at a selected speed, amplified signals generated by race roughness and dirt indicate bearing quality. *Baker Co. Inc., Maplewood, Me.*

Torsion Spring Tester: Universal tester checks loads and deflections of torsion, double torsion, spiral, clock and power springs. Meets National Bureau of Standards requirements for precision scales and is accurate within 0.25 per cent. Torsion springs with wire diameter from 0.005 to 0.125-in. and OD from 1/16 to 6 in., in lengths from 1/16 to 6 in., can be tested for torque from 1/4 oz-in. to 48 lb-in. Fifteen arbors ranging in size from 1/16 to 1/2-in., block of weights ranging from 1/4 oz-in. to 4 lb-in., five 8-lb-in. weights and two adapters for extremely small springs are provided. Instrument is 9 in. high, 11 in. wide, 15¾ in. long. *Carlson Co., New York, N. Y.*

MEN OF MACHINES

Roger W. Bolz

Penton Publishing Co. has announced the appointment of **Roger W. Bolz** as Editor of its new magazine, *Automation*. Widely known as an authority on designing for production, Mr. Bolz has served on the staff of MACHINE DESIGN since December 1944. He brings to *Automation* — which will be devoted to management, economics, operation and design of automatic machines, automated lines, complete systems and plants—a wealth of experience and background.

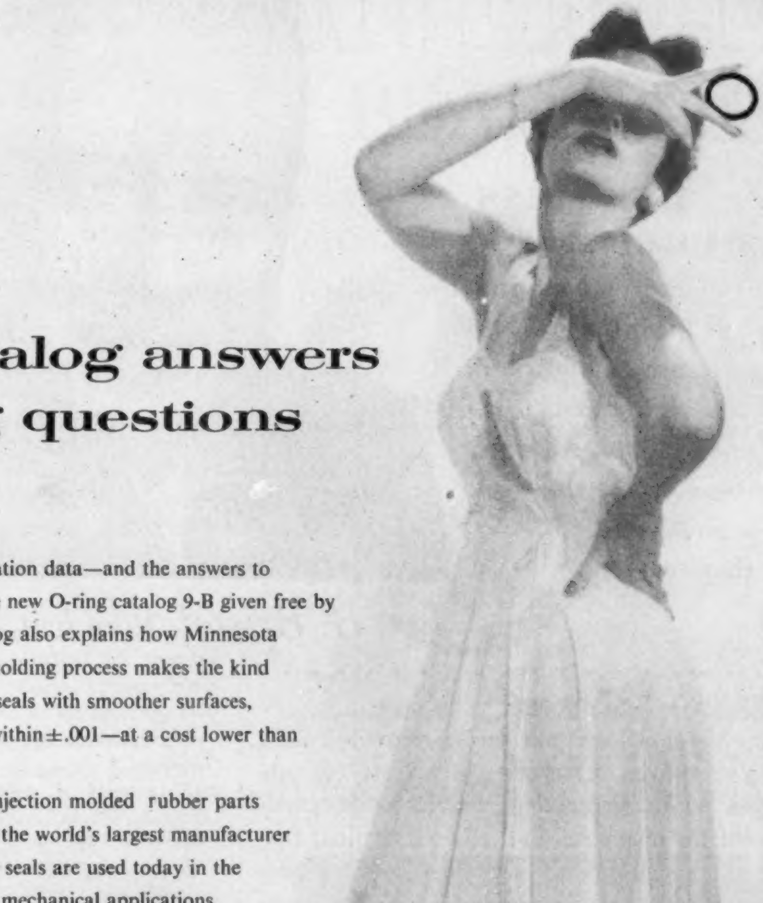
A registered professional engineer (Ohio), Mr. Bolz received his engineering education at Case Institute of Technology. Practical production and design experience with such firms as Great Lakes Aircraft Co., Weatherhead Co. and Reliance Electric & Engineering Co. led to his position as assistant works engineer for National Carbon Co., where he had charge of machine design and development, production design, product quality and tooling, and general plant equipment maintenance. Early experience as assistant to R. E. Sheal, consulting engineer on materials handling equipment, and with the Stephens-Adamson Co. in continuous processing plant layout has provided him with a keen appreciation of the problems and opportunities in the growing field to be served by *Automation*.

A member of the American Society of Mechanical Engineers, Mr. Bolz serves on the executive committee of the Machine Design division. He is also a member of the National Society of Professional Engineers and the Cleveland Engineering Society.

Author of many articles on design for production and producibility, Mr. Bolz wrote the two-volume book, *Production Processes—Their Influence on Design*, is a contributor to *Kent's Mechanical Engineers' Handbook* and is editor of the new *ASME Handbook—Processes*.

♦

Sealtron Corp., Cincinnati, has announced the appointment of **Franklin R. Kadison** as director of research and development. Mr. Kadison comes to Seal-



New catalog answers O-ring questions

You'll find complete installation data—and the answers to most O-ring questions—in the new O-ring catalog 9-B given free by Minnesota Rubber. The catalog also explains how Minnesota Rubber's exclusive injection molding process makes the kind of ring seals you need today: seals with smoother surfaces, tougher structure, tolerences within ±.001—at a cost lower than the industry average.

World-wide acceptance of injection molded rubber parts now make Minnesota Rubber the world's largest manufacturer of O-rings. Minnesota Rubber seals are used today in the finest hydraulic, electrical and mechanical applications.

No matter how tough your O-ring problem, rest assured that Minnesota Rubber will find the best answer at the lowest possible cost. Write today for complete details. Don't forget to request your free catalog 9-B, "O-rings," for complete installation data.

Minnesota Rubber and Gasket Company

3630 Wooddale Avenue, Minneapolis 16, Minn.
Dept. 302
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Men of Machines

tron from Electrical Industries of Newark, N. J., where he was assistant to the chief engineer in the hermetic seal division.

♦

John B. Holt

A newcomer to the staff of MACHINE DESIGN is Assistant Editor **John B. Holt.** Mr. Holt is introduced in this month's "Over the Board" columns, Page 4.

♦

President and a director of North American Aviation Inc., **John Leland Atwood** has been elected president of the Institute of the Aeronautical Sciences for 1954. He is a Fellow of the Institute. Four newly elected vice presidents are **William A. M. Burden,** William A. M. Burden and Co.; **E. S. Thompson,** Aircraft Gas Turbine Div. of General Electric Co.; **Edmund T. Price,** Solar Aircraft Co.; and **John W. Larson,** Fort Worth, Tex., Div. of Consolidated Vultee Aircraft Corp.

♦

Paul H. Kemmer

Col. Paul H. Kemmer, USAF (Ret.), has been appointed to the newly created post of chief designer in the engineering division of Ryan Aeronautical Co., San Diego, Calif. He will be responsible for design aspects of the company's engineering projects and will originate long-range plans for future technical activities under the supervision of the director of engineering.

After studying mechanical engineering at the University of Michigan, Col. Kemmer was a flying cadet in the aviation section of the Signal Corps. He won his wings in 1919. During the next two years, as a civilian, he served as designer for the Winton and Packard motor car companies, the Dayton Wright Airplane Co. and Westinghouse Electric Corp. After service as a civilian aeronautical engineer, he returned

to active Air Force duty until his retirement five years ago. Much of Col. Kemmer's military aviation activity dealt with research, development and engineering to improve aeronautical equipment. While in service, he received the master of science degree in aeronautical engineering from the California Institute of Technology.

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William R. Hummel

William R. Hummel was recently promoted to chief engineer of Trailmobile Inc., Cincinnati. Mr. Hummel joined the company in 1930, after studying engineering at the University of Cincinnati Evening College. During World War II his work included designing special trailers and the tools necessary to build them. In 1947 he was promoted from assistant production engineer to development engineer in charge of the experimental and development department. Three years later he was named executive engineer, the post he leaves to assume his new duties.

•

Appointment of **A. R. Ficker** as director of advanced engineering for the Rockford Clutch Div. has been announced by Borg-Warner Corp., Rockford, Ill. Mr. Ficker will be responsible for the development of new product lines. He was recently associated with Chicago Midway Laboratories, a division of the University of Chicago, as acting chief engineer and consulting engineer on a special Air Force project, and before that was a staff engineer at Willys-Overland Motors Inc.

•

Robert A. Wagner has been appointed chief engineer of Hiller Helicopters, Palo Alto, Calif. He was formerly chief engineer of the Aircraft Div. of McCulloch Motors.

•

Appointment of **John E. Brennan** as general manager was recently announced by the newly formed Automotive Body Div. of Chrysler Corp., Detroit. Mr. Brennan joined the company as a graduate student in the Chrysler Institute of Engineering in 1934. He worked in various departments of the central engineering division, in the resident engineer's office at the Plymouth plant, and in 1942 was appointed chief production engineer at the Dodge-Chicago plant. Following World War II, he returned to Detroit in the

engineering department of the Chrysler-Jefferson plant. In 1946 he was named assistant chief engineer at the Dodge main plant and three years later was assigned to the staff of the operating manager of the Dodge Div. His next assignment was that of general manager of the Chrysler Jet Engine Plant, the position he held until his present appointment.

•

Eldred P. Codling

The Hoover Co. has appointed **Eldred P. Codling** chief engineer of its Electric Motor Div. Mr. Codling has had 20 years of experience in various design and executive engineering capacities with Westinghouse Electric Corp., Delco Products Div. of General Motors Corp., and Jack & Heintz Inc. His new work will involve all types of fractional and integral horsepower electric motor manufacture, including power plants for many domestic appliances and industrial machine tools. He will work closely on the development of special electrical and mechanical details for custom-built motors and will also be concerned with the production of electric motors for the parent firm's production of vacuum cleaners and other home appliances.

•

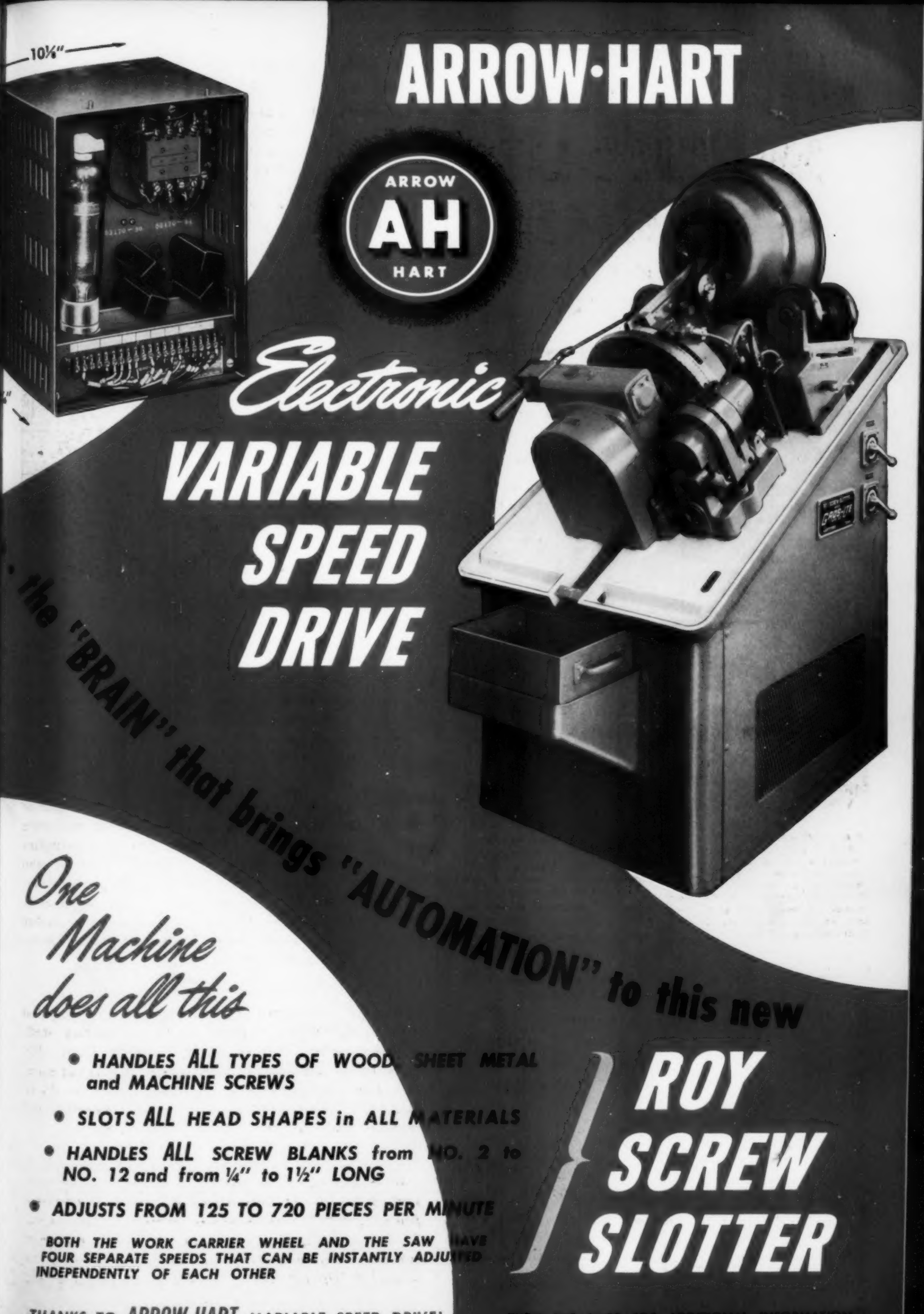
Robert G. Dobbin was recently named manager of the commercial engineering section of Jack & Heintz Inc., Cleveland. He will direct engineering activities on fractional-horsepower motors. Mr. Dobbin, who joined the company a year ago as a design engineer, previously was a designer of fractional-horsepower motors with General Electric Co. and acted as motor consultant for International General Electric at Schenectady, N. Y., and in Brazil.

•

Wheeler Associates Inc., Cleveland, has appointed **Robert Nightingale** director of its engineering staff. Formerly chief engineer of the Colson Corp., Mr. Nightingale has also been associated in management and supervision of engineering design with the Glenn L. Martin Co. Inc., Firestone Tire and Rubber Co. and Great Lakes Aviation Corp.

•

D. W. Brown will fill the newly created post of manager of project engineering in the piloted aircraft engineering division at Goodyear Aircraft Corp., Akron, O., and **S. J. Pipitone** will succeed him as manager of airframe installations design. Mr. Brown joined the Goodyear Zeppelin Corp. in 1929 as a draftsman and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... the "BRAIN" that brings "AUTOMATION" to this new
ROY
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does all this
• HANDLES ALL TYPES OF WOOD, SHEET METAL and MACHINE SCREWS
• SLOTS ALL HEAD SHAPES in ALL MATERIALS
• HANDLES ALL SCREW BLANKS from NO. 2 to NO. 12 and from ¼" to 1½" LONG
• ADJUSTS FROM 125 TO 720 PIECES PER MINUTE
BOTH THE WORK CARRIER WHEEL AND THE SAW HAVE FOUR SEPARATE SPEEDS THAT CAN BE INSTANTLY ADJUSTED INDEPENDENTLY OF EACH OTHER
THANKS TO ARROW-HART VARIABLE SPEED DRIVE!
TURN THE PAGE FOR ADDITIONAL INFORMATION

began doing design work in 1931. During World War II he managed Goodyear Aircraft's structural design operations. Mr. Pipitone, who joined the company last September, was formerly associated with the Glenn L. Martin Co., Canadair Ltd. and Chance Vought Aircraft Co.

♦

L. A. Kilgore has been appointed staff engineering manager for the East Pittsburgh divisions of Westinghouse Electric Corp. He will direct the engineering laboratories and assist in the direction of design and development of the electrical generating and distribution apparatus produced at the plant.

♦

William R. Hough

William R. Hough, engineering vice president of Reliance Electric & Engineering Co., Cleveland, was elected a director of the company at its annual meeting of shareholders. Mr. Hough joined the company upon graduation from the University of Michigan in 1929, then held several positions in the engineering organization which led to his election as engineering vice president in 1948. He is a member of the American Institute of Electrical Engineers, the Association of Iron and Steel Engineers, the Cleveland Engineering Society and the Cleveland Technical Societies Council.

♦

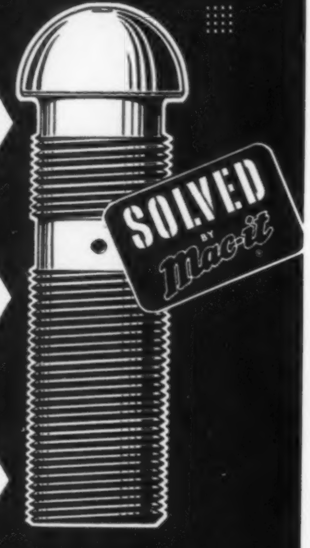
Dudley D. Fuller has been appointed principal scientist on the staff of the mechanical engineering division of the Franklin Institute's Laboratories for Research and Development, Philadelphia. He will continue his work as associate professor of mechanical engineering at Columbia University. Author of many technical articles in the field of mechanical engineering, Mr. Fuller wrote a four-part series entitled "Hydrostatic Lubrication," which appeared in MACHINE DESIGN in 1947.

♦

With duties which include engineering for machine development and instruments and controls, **John W. Pearson** has been promoted to executive engineer in charge of engineering and research development by Minnesota Mining & Mfg. Co., St. Paul, Minn.

♦

W. C. O'Connell has been appointed general manager of the aircraft accessory turbine department of



Men of Machines

General Electric Co. at Lynn, Mass. He has named **W. O. Meckley** manager of engineering for the department.

♦

The Permutit Co., New York, has promoted **Felix W. Saco** to the position of development engineer. When he joined the company in September 1952, Mr. Saco was assigned to the mechanical engineering department.

♦

The appointment of **Robert Von Rotz** as chief engineer has been announced by Tuthill Pump Co., Chicago. His 27 years of technical and management experience in engineering have trained Mr. Von Rotz in research, development and design in both low and high-pressure pumps. He has been associated with Ingersoll Milling Machine Co., Vickers Inc. and the New York Air Brake Co. Just prior to joining Tuthill, he served as chief engineer of Applied Research and Development Corp. of New York City and Berne, Switzerland.


Robert Von Rotz

♦

M. H. Hobbs has been appointed manager of the Switchgear Div. of Westinghouse Electric Corp. Having joined the division in 1922, he was named manager of the switchgear engineering department in 1944 and assistant manager of the division in March 1953.

♦

Associated with the company since 1933, **C. R. Thorpe** has been named chief engineer of H. H. Buggie Inc., Toledo, O.

♦

Carl Hammon has joined the engineering staff of the hydraulic press division of Erie Foundry Co., Erie, Pa.

♦

J. A. Zurn Mfg. Co., Erie, Pa., recently appointed **Albert A. Baker** as executive assistant and **John L. Derby** as plant manager.

♦

Beaver Precision Products, Clawson, Mich., has appointed **Richard E. Sears** to the post of vice president in charge of engineering. Prior to joining the company, in 1950, he was associated with Jack & Heintz Inc. in several engineering capacities, most recently as staff mechanical engineer.

THE ENGINEER'S Library

Recent Books

Kinematics of Mechanisms. *By N. Rosenauer and A. H. Willis, New South Wales University of Technology, Australia; published by Lange, Maxwell & Springer Inc., New York; available from* MACHINE DESIGN, *$9.85 postpaid.*

Following short introductory chapters dealing with plane motion, kinematic pairs and chains, the next two sections cover the Euler-Savary equation and its graphical presentation, and the Bobillier constructions. A major portion of the material is devoted to a discussion of straight-line motion, relative motion of three or four planes, kinematic chains of *n* links, and mathematical and graphical velocity and acceleration determinations and constructions. The final chapter of the book gives an introduction to the synthesis of mechanisms and a five-page appendix covers Coriolis acceleration.

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Electrical Engineering. *By Fred H. Pumphrey, professor of electrical engineering, University of Florida; 416 pages, 6 by 9 inches, clothbound; published by Prentice-Hall Inc., New York; available from* MACHINE DESIGN, *$8.00 postpaid.*

A second edition, this book presents fundamental electrical engineering theory along with applications of that theory in commercial practice. The first twelve chapters review basic theory of dc and ac measurements, circuits, generators, and motors. Chapter thirteen covers electric motor characteristics, housings, ratings, control, and selection. The final chapters include material on electron tubes and circuits; heating, welding, and electrochemical processes; electrical illumination; industrial electrical measurements; industrial wiring systems; and electric power and its economics and maintenance.

Manufacturers' Publications

Selenium Rectifier Handbook. 80 *pages,* 5¼ *by* 8½ *inches, paperbound; available from Federal Telephone and Radio Co., Selenium-Intelin Dept.,* 100 *Kingsland Rd., Clifton, N. J.,* 50 *cents per copy.*

Design, specification and circuitry of selenium rectifiers are given detailed descriptive treatment in this handbook for radio, television, and other electronic applications. Not only are available selenium rectifiers listed, but the booklet also covers power supply circuits for such applications as phonographs,

audio and photocell amplifiers, radios, intercommunication systems and other dc power supply requirements.

New Codes and Standards

Gas Turbine Power Plants. *PTC* 22-1953; 25 *pages, 8½ by* 11 *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W.* 39*th St., New York* 18, *N. Y.,* $1.50 *per copy.*

Methods for conducting commercial, routine, and acceptance tests on gas turbine power plants and components are specified in this code. Descriptions and definitions of terms, instruments and methods of measurement, and computation of results for gas turbine power plants are included in this booklet.

♦

Buttress Screw Threads. *ASA B*1.9-1953; 17 *pages, 8½ by* 11 *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W.* 39*th St., New York* 18, *N. Y.,* $1.00 *per copy.*

This standard covers screw threads of the buttress form and gives drawings of a standard form of such threads. Tables of preferred diameters and preferred pitches along with a formula for calculating effective pitch diameter tolerances are included. Additionally, tolerances for major and minor diameters, a system of allowances between mating parts, and recommended methods of gaging are given.

♦

Steel Pipe Flanges and Flanged Fittings. *ASA B*16.5-1953; 75 *pages, 8½ by* 11 *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W.* 39*th St., New York* 18, *N. Y.,* $3.00 *per copy.*

Cast and forged steel flanges, cast and forged steel flanged fittings, and cast and forged steel flanged and butt-welded end valves are embraced in this standard. Known as the American 150, 300, 400, 600, 900, 1500 and 2500-pound steel flange standards, this reference covers pressure ratings, sizes and methods of designating openings, marking, minimum requirements for materials, dimensions, tolerances, and tests.

Government Publications

NACA Technical Series. *Each publication is* 8 *by* 10½ *inches, paperbound, side-stapled; copies available from National Advisory Committee for Aeronautics,* 1924 *F St. N. W., Washington* 25, *D. C.*

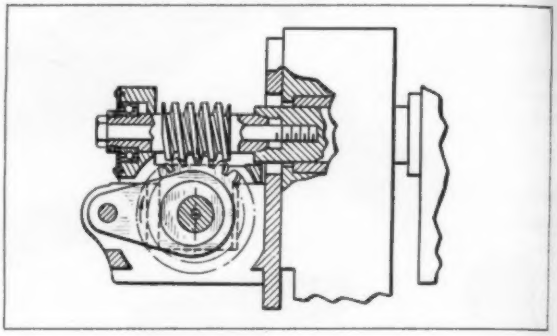
The following Technical Notes are available:

3029. **A Fundamental Investigation of Fretting Corrosion**—52 pages.
3039. **Experimental Stress Analysis of Stiffened Cylinders with Cutouts—Pure Torsion**—41 pages.
3043. **Application of Silver Chloride in Investigations of Elasto-Plastic States of Stress**—55 pages.

NOTEWORTHY

Patents

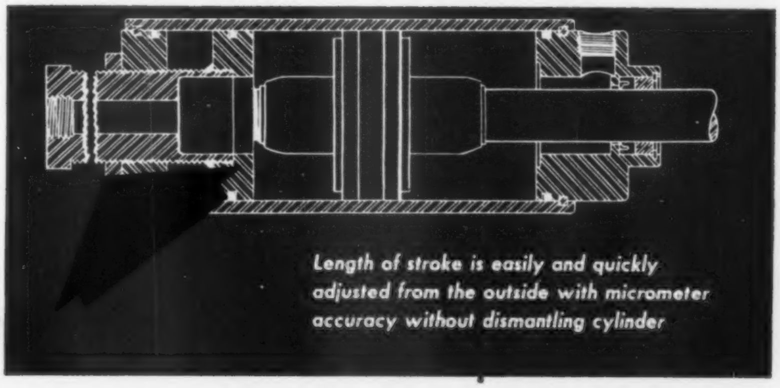
RECIPROCATING LINEAR MOTION of rotating shafts at a controlled rate is afforded by a unique worm and gear mechanism designed by Henry E. Willitts and assigned to H. B. Smith Machine Co. Covered in patent 2,646,685, the device consists of a worm and gear set attached to the end of the rotating shaft and mounted in a carriage free to slide in a fixed frame. Rate of oscillation is a function of the gear reduction ratio. Reciprocating action is

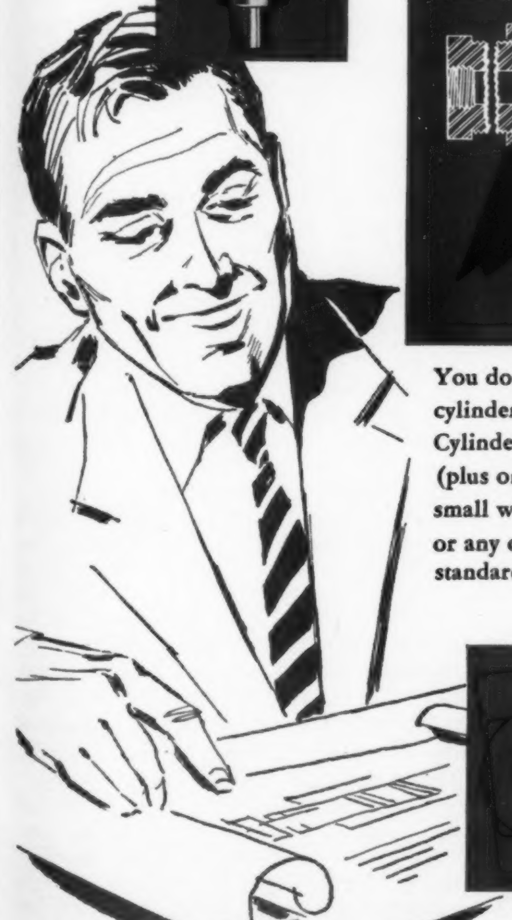


produced by hubs, mounted eccentrically on the shaft carrying the gear, which are journalled in arms connected to the fixed frame. In operation, as the worm turns with the main shaft and drives the gear and its shaft, the arms act as connecting rods to drive the carriage and main shaft back and forth. A smooth oscillating action is achieved by the design which transmits gear loads to the fixed frame and minimizes the possibility of wear and chattering of the moving parts.

TORSIONAL LOCKING of antifriction bearings by means of a resilient support mounting permits axial movements under load but prevents rotation of the bearing outer race. Assigned to Bendix Aviation Corp. by Giltner J. Knudson, the support mounting shown in patent 2,650,866 employs a rubber ring frictionally mounted to the outer race to suspend the bearing in an oversize circular opening in the housing. Engagement of the rubber ring and race surfaces is maintained under pressure by a sinuous metal spring ribbon, mounted on the periphery of the ring, which fits into a circular slot in the housing and acts to hold the ring stationary. Uniform friction forces produced at the ring contact surface prevent rotation of the bearing race but permit axial movements under load. In addition, the design eliminates the need for close assembly tolerances and compensates auto-

standard O-M cylinders are easily and quickly modified to fit SPECIAL APPLICATIONS





You don't pay for special patterns . . . and you don't have to use a cylinder that's not quite "right" for the job when you choose the O-M Cylinder. Since nearly all parts are standardized in O-M Cylinders (plus one or two semi-standard parts), the additional cost is extremely small when you need a duplex, triplex, adjustable-stroke, tandem— or any other "special" type cylinder for applications not suited to standard units.

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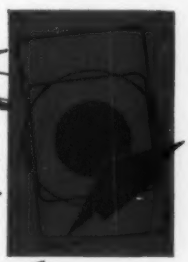

O-M Cylinders require ⅓ *less installation space* than conventional cylinders of the same bore because O-M's SPECIAL INTERLOCKING MECHANISM does away with projecting tie rods and end caps. Also provides better balance, reduces distortion . . . giving O-M the *lowest coefficient of friction of any cylinder.* End plugs tapped for universal mounting. Any one or combination of mounting brackets may be used to install without disassembling or changing cylinder. Easily removed, inspected, repaired. ALL-STEEL with bearing bronze—no castings.

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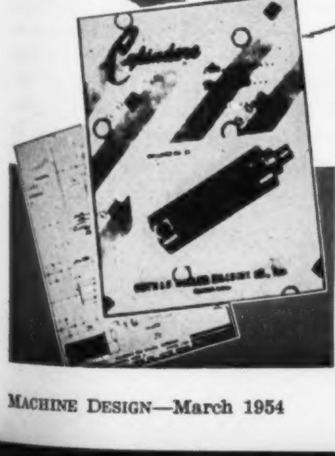

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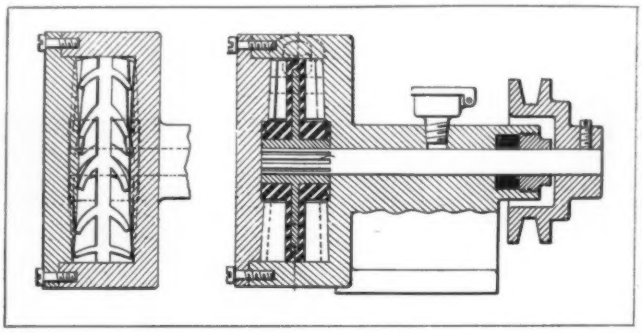
matically for variations in clearance between mounted parts. A modified unit for use on the bearing inner race is also shown in the patent.

HERRINGBONE VANES of a flexible material are utilized to create pumping action in a rotary pump design detailed in patent 2,649,052. The pump has a rubber rotor mounted in a fluid chamber enclosed by two converging endwalls. Pumping and suction actions are produced by V-shaped radial vanes which flex in and out as they rotate between the inclined walls.



Fluid intake and delivery ports are located on the circumference of the pumping chamber. Suitable for high-speed operation, the design can also be readily adapted for use as a hydraulic motor. A modified rotor design for operation in both directions of rotation is also shown in the patent which has been assigned to Marine Products Co. by Ferdinand M. Weyer.

STANDBY SEALING of rotating or reciprocating shafts to facilitate maintenance in service is accomplished with an inflatable seal unit assigned to Syntron Co. and described in patent 2,648,554. Designed for use with a stationary housing, the seal is utilized as a temporary barrier to relieve fluid or lubricant pressure on the main sealing unit while repairs are being made. Sealing element is an inflatable tube, similar to the familiar inner tube, rigidly fastened to the housing. Expansion of the tube, which may be accomplished by pneumatic, hydraulic or mechanical means, forces a wear ring on the inner tube surface into sealing engagement and prevents fluid flow along the shaft; the degree of sealing is controlled by the amount of tube expansion. Deflation of the tube permits the main sealing unit to return to operation. Several modifications of the design for different degrees of sealing engagement are shown in the patent which has been assigned by inventor Walter J. Gilbert Sr.

Copies of the patents briefed in this department may be obtained for 25 cents each from The Commissioner of Patents, Washington 25, D. C.